Definition of An Improved Wet Support Bridge Concept And Related System Analysis

Report to
US Army Mobility Equipment Research
and Development Command (MERADCOM)

Contract No. DAAK70-79-D-0036
Task Order No. 0019

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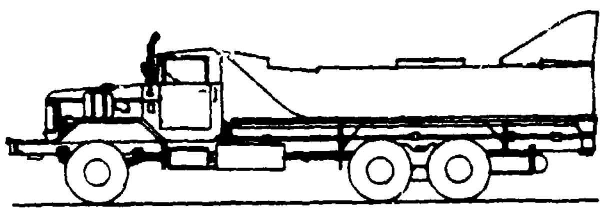

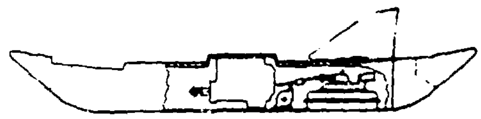

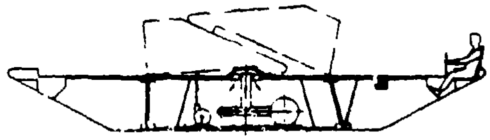

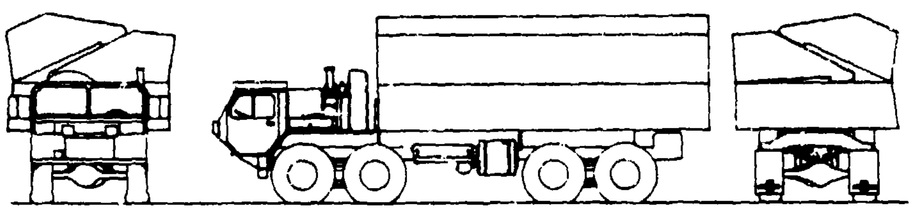

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This study presents key findings for the concept of an Improved Wet Support Bridge based on integral propulsion in all bays. It also provides a new concept for an integral propulsion bay for the existing Ribbon Bridge. The study concluded that a new half-width interior bay with integral propulsion in a single folding bow ponton is practicable for the existing Ribbon Bridge system and will improve bridge/rafting performance and reduce life cycle costs in comparison with the Current Ribbon Bridge. It also concluded that the Improved Wet Bridge System based on integral propulsion in all three-ponton bays provides

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20. Abstract (Continued)

bridge/rafting capabilities for improved performance in handling Military Load Class 70. These new concepts will provide improved bridge/rafting performance throughout the 1980s and 1990s.

DEFINITION OF AN IMPROVED WET SUPPORT BRIDGE CONCEPT
AND RELATED SYSTEM ANALYSIS

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FOREWORD

This document summarizes key findings and presents the background material relevant to the study entitled, "Definition of an Improved Wet Support Bridge Concept and Related System Analysis." This report presents a systems-analysis approach and evaluation of:

- Integral Propulsion Subsystem Survey and Analysis (Chapter 2., Phase I).
- Ribbon Bridge with Integral Propulsion (Chapter 3., Phase I).
- Three-Ponton Bay System with Integral Propulsion (Chapter 4., Phase I).
- Control System (Chapter 5., Phase I).
- Organizational and Life Cycle Costs of Alternative, Viable Wet Bridge Systems (Chapter 6., Phase II).

This study concluded that a new half-width interior bay with integral propulsion and a single folding bow ponton is practicable for the existing Ribbon Bridge System and will improve bridge/rafting performance and reduce life-cycle costs in comparison with the Current Ribbon Bridge. It also concluded that the Improved Wet Bridge System based upon integral propulsion in all three-ponton bays provides bridge/rafting capabilities for improved performance in handling Military Load Class 70.

The new half-width interior bay with integral propulsion would provide improved performance for the Ribbon Bridge System throughout the late 1980s and into the 1990s. Beginning in the early 1990s, the integral propulsion three-ponton bay, a totally new Wet Bridge concept, would provide improved bridge/rafting performance.

This report is submitted to the US Army Mobility Equipment Research and Development Command (MERADCOM), Fort Belvoir, Virginia 22060 by Arthur D. Little, Inc., 20 Acorn Park, Cambridge, Massachusetts 02140, and was prepared under Task Order No. 0019 of Contract No. DAAK-70-79-D-0036. It was prepared under the guidance of Messrs. Jerry Dean, Carlos Piad, and Martin E. Falk of MERADCOM. Questions of a technical nature should be addressed to Robert H. Bode, 617-864-5770, the Manager of the study and principal investigator. Other investigators included John S. Howland, Edward G. Pollak, Robert M. Lucas, Nicholas A. Memmo, and William F. Reehl under the administrative cognizance of Roger G. Long.

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1. SUMMARY

1.1 INTRODUCTION

This report presents the results of a system analysis and concept formulation for incorporating integral propulsion into the Improved Wet Support Bridge--both the existing Ribbon Bridge System and the new Three-Ponton Bay System designed for Military Load Class 70 Tracked and Wheeled Vehicles. The study was conducted in two phases:

Phase I - Formulation and Presentation of Systems for Incorporating Integral Propulsion into the Improved Wet Support Bridge

Phase II - Organizational and Life Cycle Cost Aspects of the Comparative Systems Concepts

1.2 BACKGROUND

The US Army needs a cost-effective Improved Wet Support Bridge System to enhance its wet gap crossing capabilities and readiness beyond 1985. The purpose of this task is to evaluate the potential of using integral propulsion subsystems instead of bridge erection boats to power the wet support bridge modules. This continuous float bridge is planned to be in the US Army inventory in the 1990s.

1.3 OBJECTIVE

The original objective of Phase I was to formulate and present system concepts for incorporating integral propulsion into the Improved Wet Support Bridge. This objective was amended early in the program to include conceptual definitions of integral propulsion in:

a. The Ribbon Bridge

(1) With pump-jet

(2) With outboard drive

b. The Three-Ponton Modular System (as described by Figures 1, 2, and 3 of Statement of Work).

(1) With pump-jet

(2) With outboard drive

The objective of Phase II was to develop organizational and life-cycle costs for the alternative bridge systems, namely,

- Current Ribbon Bridge
- Improved Ribbon Bridge with Integral Propulsion Half-Bays
- Improved Wet Bridge with Integral Propulsion Three-Ponton Bays

The ten-year projected life-cycle costs were to include:

- Acquisition costs
- Crew costs
- Training mission operating and support costs

1.4 SCOPE OF WORK

Phase I involved four subtasks. These are listed below along with the numbers of the sections in which they are discussed.

2. Integral Propulsion Subsystem Survey and Analysis
3. Ribbon Bridge with Integral Propulsion
4. Three-Ponton Bay System with Integral Propulsion
5. Control System

The scope of work of Phase II involved a single subtask:

6. Organizational and Life-Cycle Costs of Alternative, Viable Wet Bridge Systems

1.5 FINDINGS OF PHASE I

1.5.1 Integral Propulsion Subsystem Survey and Analysis (Section 2.)

For the Ribbon Bridge interior bay, only an outboard propeller thruster was acceptable, namely, a Schottel SRP-12 unit driven by a

Garrett or Turbomach gas turbine. Unless variable-speed drive is used, a special design and procurement would be required for a controllable pitch propeller.

For the preferred concepts, namely, the Improved Ribbon Bridge with Integral Propulsion Half-Bays and the Improved Wet Bridge with Integral Propulsion Three-Ponton Bays, the optimum thrusters and associated prime movers are summarized below. More detailed information is presented in the Propulsion Equipment Summary, Table 1-1.

Bridge/Bay	Thruster	Prime Mover
Improved Ribbon Bridge Integral Propulsion Half-Bay ("Slice")	SPJ-50	BF6L 913
Improved Wet Bridge/ Integral Propulsion Three-Ponton Ramp Bay	SPJ-32	F6L 912
Improved Wet Bridge/ Integral Propulsion Three-Ponton Interior Bay		
Greater than 1800 lb thrust per bay up to 3000 lb/bay	SPJ-32	F6L 912
1800 lb thrust and less per bay	SPJ-20	F4L 912

1.5.2 Study of Ribbon Bridge with Integral Propulsion (Section 3.)

No acceptable way was found to install the Schottel Pump-Jet and its prime mover in either the interior or ramp bay of the Current Ribbon Bridge.

An outboard drive can be installed in the bow pontons of the Current Ribbon Bridge interior bay, but it will reduce the bridge's already marginal buoyancy.

Table 1-1

Propulsion Equipment Summary

	Thruster			Prime Mover		
Model No.	Maximum Thrust (lb)	Input Speed (rpm)	Hull Opening Required (dia., in.)	Engine Model No.	Maximum DIN Horsepower	Output Speed (rpm)
SPJ-50	2,248	2,300	47.2	Deutz BF6L 913	160	2,800
SPJ-32	1,574	2,800	43.3	Deutz F6L 912	110	2,800
SPJ-20	966	2,800	34.0	Deutz F4L 912	73	2,500

The preferred concept, which provides integral propulsion both for the Ribbon Bridge and for rafting, is the new integral propulsion half-bay* with a single folding bow ponton. This concept results in an Improved Ribbon Bridge System with greater bridging and rafting capabilities.

The Improved Ribbon Bridge integral propulsion half-bay with a single folding bow ponton would be carried on the present Ribbon Bridge transporter (CONDEC Model No. 2280). The deadweight of the integral propulsion half-bay having a propulsion subsystem based upon two SPJ-50 Pump-Jets and two six-cylinder air-cooled turbocharged Deutz diesels would be 6.00 ST.

1.5.3 Study of Three-Ponton Bay System with Integral Propulsion (Section 4.)

The preferred concept for a propulsion subsystem in a three-part interior bay is a Schottel Pump-Jet clutched to a Deutz air-cooled diesel. The diesel may have either four or six cylinders, depending upon the thrust needed. The Level 1 drawings are based upon the six-cylinder engine.

The deadweight of the three-part interior bay with integral propulsion based upon the pump-jet is estimated to be 8.94 ST, resulting in a capacity of 34.14 ST with the roadways awash.

Because of space limitations, only a single pump-jet propulsion subsystem can be installed in the roadway portion of the three-part ramp bay.

An outboard drive can be installed in the bow pontons of the three-part interior bay to provide integral propulsion utilizing a gas turbine as the prime mover. However, the outboard drive is considered more complicated, less reliable, and more vulnerable than the pump-jet drive for this application.

*This concept is referred to by MERADCOM as "The Slice".

The bow pontons are stowed and deployed by means of a multiple-linkage hydraulic system. They pivot on "invisible" hinges that do not protrude above the roadway when the pontons are deployed.

Both the interior and ramp bays would be carried on a transporter chassis, based upon the new HEMTT 10-ton, 8 x 8 truck chassis. The Heavy Expanded Mobility Tactical Truck family is in a prototype procurement phase with Oshkosh Truck Corporation.

1.5.4 Study of the Control Subsystem (Section 5.)

The two propulsors of each module (bay) are ganged both in power and direction in order to provide a single thrust vector for control purposes. Each module can be manually controlled individually or can be ganged for multiple module control.

Up to four integral propulsion modules (bays) can be controlled from a portable console, which can be emplaced in either of the two control stations on any of the four bays.

The communication link between modules is hardwired through a retractable reel system in each module.

The integrated control system has no automatic feedback loop; the only feedback is provided by the operator via visual references or directional sensors.

1.6 FINDINGS OF PHASE II

1.6.1 Organizational and Life-Cycle Costs of Alternative, Viable Wet Bridge Systems (Section 6.)

The acquisition cost, including spares, for each of the alternative bridge systems, based upon bridging a 120-m wet gap, is as follows:

Bridge System	Acquisition Cost FY '82 ($000)
Current Ribbon Bridge	4,139
Improved Ribbon Bridge with Integral Propulsion Half-Bays	3,307
Improved Wet Bridge with Integral Propulsion Three-Ponton Bays	4,891

The ten-year life-cycle crew cost for these bridge systems is as follows:

Bridge System	Ten-Year Life Cycle Crew Cost FY '82 ($000)
Current Ribbon Bridge	8,455
Improved Ribbon Bridge with Integral Propulsion Half-Bays	7,322
Improved Wet Bridge with Integral Propulsion Three-Ponton Bays	6,955

The ten-year life-cycle operating and support costs, based upon one training mission per month, are as follows:

Bridge System	Operating and Support Costs Based Upon One Training Mission/Month FY '82 ($000)
Current Ribbon Bridge	899
Improved Wet Bridge with Integral Propulsion Half-Bays	709
Improved Wet Bridge with Integral Propulsion Three-Ponton Bays	1,067

The total ten-year life-cycle costs based upon the above three cost elements are as follows for each of the alternative bridge systems:

Bridge System	Limited Life-Cycle Costs Based Upon Acquisition, Crew, and Mission Cost Elements FY '82 ($000)
Current Ribbon bridge	13,493
Improved Ribbon Bridge with Integral Propulsion Half-Bays	11,338
Improved Wet Bridge with Integral Propulsion Three-Ponton Bays	12,913

1.7 CONCLUSION

Integral propulsion of the half-width interior bay with a single folding bow ponton is practicable for the Ribbon Bridge system and will result in improved bridge/rafting performance at lower life cycle cost than with the current Ribbon Bridge.

The Improved Wet Bridge system, based upon integral propulsion in all three-ponton bays, provides bridge/rafting capabilities for improved performance in handling Military Load Class 70. Dual integral propulsion can be provided in the interior bay utilizing pump-jet thrusters. Single pump-jet integral propulsion can be provided in the ramp bay. The estimated life cycle cost of the Improved Wet Bridge system with ample Military Load Class 70 performance and improved land mobility is less than that of the current Ribbon Bridge system.

1.8 RECOMMENDATIONS

To meet Army needs for increased bridge/rafting performance in the 1980s and into the early 1990s, it is recommended that the Improved Ribbon Bridge with half-width interior bays and single folding bow pontons be developed with integral propulsion.

For the 1990s and beyond, it is recommended that an improved wet bridge/rafting system be developed with three-ponton interior and ramp bays utilizing integral pump-jet propulsion.

2. INTEGRAL PROPULSION SUBSYSTEM SURVEY AND ANALYSIS

2.1 INTRODUCTION

The purpose of this study is to identify and evaluate the various alternative components of a propulsion system for two military wet-gap bridging systems:

(a) The conventional ribbon bridge.

(b) The proposed three-part modular system.

Emphasis has been placed in this study on two types of thrusters:

(a) An integral jet pump thruster using the Schottel Pump-Jet as the baseline unit, and

(b) An outboard propellor thruster driven by an inboard engine.

Alternative thrusters falling into these generic classes have been considered as well as a number of alternative prime movers.

2.2 PROPULSION SYSTEM REQUIREMENTS

2.2.1 Performance

(a) Thrust

The basic performance requirement for the propulsion system is that it provide sufficient thrust in each bridge bay:

- To permit station-keeping in the bridge mode with Military Load Class 70 crossings in a stream with velocities up to 2.5 m/s, and
- To permit rafting with Military Load Class 70 in streams with velocities up to 3.0 m/s.

For purposes of this study, the thrust requirement was specified to be based on two specific performance levels:

- 1800 lb per 7-meter bridge module
- 1300 lb per 7-meter bridge module

In each case, of course, this thrust would have to be available at a relative stream velocity of 3 m/s to meet the above operational requirements.

(b) Direction of Thrust

In the normal operations of bridge modules, thrust may be needed in virtually any direction relative to the module itself. Thus, the thruster must be steerable. Maneuverability and response are greatest if the thruster can be steered or pointed at any angle throughout a complete 360° rotation relative to coordinates fixed to the module. However, this capability may not be required if a turning moment can be applied; the module can be rotated by use of the thrust moment until an available thrust direction is reached. Thus, the range of steerability was reserved for possible tradeoff against other desirable features.

(c) Thrust Moment

No specification has been set for the required thrust moment or, for that matter, the rate of turn of the module.

One method for setting a tentative moment specification consistent with the above thrust requirements is to assume that the bridge module is crosswise to the stream and that half of its length is in calm water or eddies associated with the shore effect while the other half is in fast water that imposes a drag consistent with the required total thrust (i.e., 650 or 900 lb on half of the hull). In this case, the reaction moment to keep the module from rotating would have to be 4530 ft-lb or 6273 ft-lb respectively, assuming a module with a length of 8.5 m (27.9 ft).

2.2.2 Configuration

(a) Thruster Location

Schematic diagrams of the cross-section (perpendicular to the bridge length) of the two types of bridge modules or bays are shown in Figure 2-1. In each case the pontons are shown in folded position for transporting and open position as used for bridging or rafting.

As the bridge bays are often operated in shallow water, the thrusters should not protrude below the bottom of the hull in the center portion of the bay. A thruster could be located along either side or at the ends of the bay so that it would be below the waterline but not protrude below the hull bottom; however, because individual bays mate with other bays along both sides to form bridges or rafts, the thruster must be located either flush with the bottom of the hull in the center portion of a bay or between the waterline and hull bottom at the bow and stern.

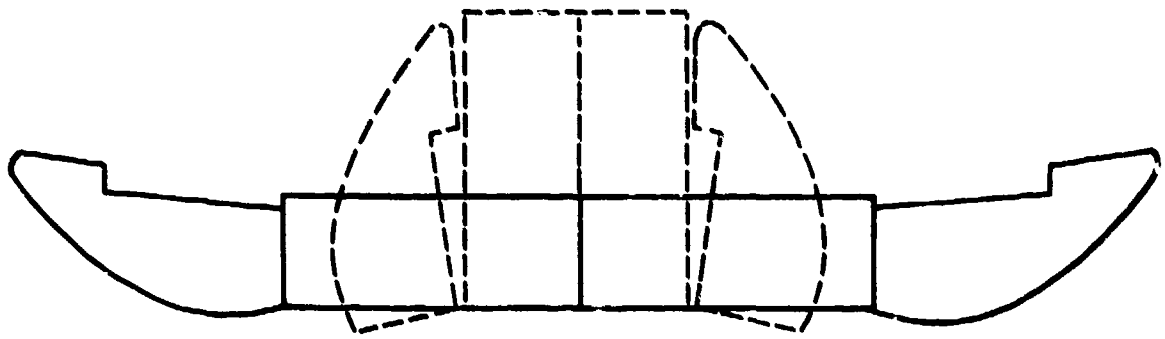
(b) Propulsion System Height

Depending on the type of bridge module, the roadway surface lies either along the outer portion of the center section(s) or the inner portion of the bow and stern sections. In these areas, the propulsion components cannot extend above the top surface of the module.

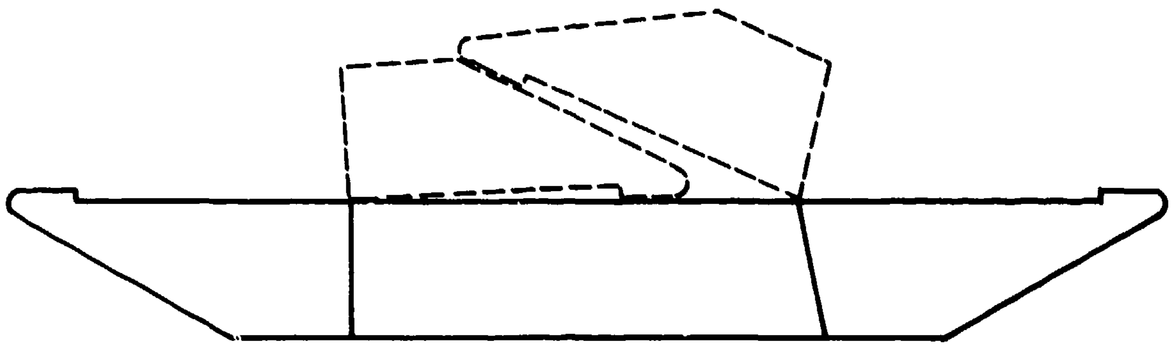
Although undesirable from the point of view of vulnerability to damage, it is conceivable that some slight extension above the roadway level could occur in the area between the roadway strips. This would be limited by the minimum clearance between tracks or wheels of all of the vehicles using the bridge.

2.2.3 Logistics

The primary logistical requirement on the propulsion system is that it should be capable of running on the fuel used by the field army of the



a. Ribbon Bridge Bay



b. Three-Ponton Bay

Figure 2-1

Schematic Diagram of Bridge Modules in Folded and Deployed Configurations future. This is, of course, diesel fuel. Thus, the prime mover must be either a diesel engine or a gas turbine. Gasoline engines were not considered as candidates for the system.

2.2.4 Maintainability

It is desirable that the propulsion system be located and mounted in such a fashion as to permit maintenance on the engine and/or thruster while the bridge bay is out of the water in storage.

It can be seen from Figure 2-1 that access to an engine installed in either bridge would not be possible in the folded (storage) configuration. Moreover, the Ribbon Bridge cannot easily be unfolded and refolded on dry land and would be impossible to open while on its carrier vehicle.

The three-part module can be unfolded on its carrier; in fact, this would probably be done prior to launching. Thus, these bays present no major difficulty to maintenance of an installed engine through an access hatch on the deck.

If a similar engine were installed in the Ribbon Bridge, the module would have to be lifted off its carrier and either deployed in water or unfolded on land. The latter option would likely require the installation of rollers or wheels on the center sections.

This difficulty with the Ribbon Bridge could be overcome by the addition of a fifth compartment to house the propulsion system. This compartment would provide flotation and access to the engines through hatches on its top surface. When the bridge module is deployed, the two center sections would rotate up around this drive compartment and enclose it in a semicircular cut-out volume designed to receive it. Thus, the engines would always be in their normal orientation and accessible for maintenance.

This concept introduces other problems, however, and is discussed in greater detail later (section 3.6).

2.3 PRIME MOVERS

2.3.1 General Discussion

As noted earlier, the gasoline engine was eliminated as the primary power source for logistical reasons, leaving the diesel and gas turbine as contenders.

With regard to diesel engines, a selection of power ratings, weights, and sizes is shown in Table 2-1. The basic tradeoff is between the air-cooled and water-cooled types. With the exception of the Volkswagen engine, the various water-cooled varieties are somewhat heavier than air-cooled engines with comparable power capacity. They are also somewhat longer, especially when the space required for the radiator or heat exchanger is included.

The critical dimensions for installation in the bridge modules is the height of the engine. This dimension is roughly the same (about 31 to 41 inches) for all of the engines shown except the four-cylinder Volkswagen engines. The latter have a height as small as 21.6 inches in the tipped version; however, their power capability is not sufficient for the Pump-Jet thruster, and their shaft speed is nearly twice that of the other engines, which would necessitate the use of a gear-box speed reduction unit.

Thus, an air-cooled diesel would be better than a water-cooled one for this application, in view of the fact that it provides comparable performance with less weight and complexity and has comparable critical dimensions. While the Deutz and Lister units do not differ significantly in most respects, the Lister units are 5 to 6 inches larger than the Deutz units in the critical height dimension.

Table 2-1

Diesel Engine Characteristics

Air Cooled		DIN Horsepower Continuous	DIN Horsepower Automotive or Intermittent	Weight	Length	Width	Height
Deutz	F4L 912	58 @ 2300	73 @ 2500	320 kg (705 lb)	901 mm (35.4")	663 mm (25.3")	840 mm (33.06")
	F5L 912	73 @ 2300	92 @ 2800	405 kg (893 lb)	945 mm (37.2")	663 mm (25.3")	880 mm (34.6")
	F6L 912	88 @ 2300	110 @ 2800	436 kg (961 lb)	1165 mm (45.9")	663 mm (25.3")	855 mm (33.8")
	BF6L 913	135 @ 2300	160 @ 2800	485 kg (1069 lb)	1123 mm (44.3")	709 mm (27.9")	921 mm (36.3")
Lister	HR4	59 @ 2200	73.5 @ 2200	432 kg (960 lb)	1013 mm (39.9")	704 mm (27.8")	1003 mm (39.5")
	HR6	88.5 @ 2200	111 @ 2200	560 kg (1245 lb)	1292 mm (50.9")	704 mm (27.8")	1003 mm (39.5")
Water Cooled							
VW 068.2, 4 cyl. (not including radiator)		46 @ 4000	51 @ 5000	129 kg (284 lb)	498 mm (19.6")	711 mm (28")	617 mm (24.3")
VW 068.3, 4 cyl. (mounts @ 50.5° angle) (not including radiator)		46 @ 4000	51 @ 5000	129 kg (284 lb)	525 mm (20.7")	785 mm (30.9)	550 mm (21.6")
VW 076.1, 6 cyl. (not including radiator)		73 @ 4000	-	191 kg (421 lb)	775 mm (30.5")	531 mm (20.9")	823 mm (32.4")
Detroit 4-53, 4 cyl. (includes radiator)		75.7 @ 2200	107 @ 2500	621 kg (1370 lb)	1524 mm (60")	737 mm (29")	1067 mm (42")
Detroit 4-53, 4 cyl. (without radiator)		-	-	572 kg (1260 lb)	991 mm (39")	762 mm (30")	991 mm (39")
Detroit 6V-53, 6 cyl. (without radiator)		122 @ 2400	176 @ 2800	674 kg (1485 lb)	991 mm (39")	1016 mm (40")	940 mm (37")
Isuzu 4 bbl, 4 cyl. (without radiator)		65 @ 2800	72 @ 2800	315 kg (695 lb)	896 mm (35.3")	621 mm (24.4")	773 mm (30.4")
Isuzu 6 bbl, 6 cyl. (without radiator)		100 @ 2800	112 @ 2800	450 kg (992 lb)	1152 mm (45.4")	645 mm (25.4")	868 mm (34.2")
Caterpillar 3304 Marine, 4 cyl. (without heat exchanger)		75 @ 2000	88 @ 2200	830 kg (1835 lb)	1530 mm (60")	910 mm (36")	1050 mm (41")

In comparison with diesels, the gas turbine's major generic characteristics are low weight and very high shaft speed. However, for this application, its most important characteristic is that it has no crankcase or cylinder head. This suggests that it may be more amenable to storage on its side or upside down, as would be required for certain engine mounting locations. If a normal diesel is tipped too far, the crankcase oil leaks into the tops of the cylinders; the engine cannot then be operated until the combustion chambers are drained.

2.3.2 Air Cooled Diesel Characteristics

The Deutz line of air-cooled diesel engines is a promising prime power source for the bridge modules. The overall dimensions of the four-, five-, and six-cylinder models were given in Table 2-1. More detailed specifications are given in Table 2-2 for the sizes of interest. Engine dimensions are shown in Figure 2-2. Photographs of the four-cylinder engine from both sides are reproduced in Figure 2-3, and a cutaway drawing of both sides is shown in Figure 2-4.

The principal alternative to the Deutz is the Lister diesel line. Again, a full line is available from two cylinders up to six. Dimensions of the four- and six-cylinder versions were shown in Table 2-1. Table 2-3 gives the specifications for the entire line from the two-cylinder engine to the turbocharged six-cylinder engine. The corresponding power curves are shown in Figure 2-5.

2.3.3 Gas Turbine Characteristics

The principal manufacturers of gas turbines in the power range of interest are the Garrett Corporation and Turbomach Division of Solar Turbines International. Both firms were contacted to determine which models of small turbines would be most suitable for this application.

Table 2-2

Deutz Diesel Specifications

Type		F4L 912								F5L 912							
No. of cylinders		4								5							
Rotation (facing flywheel)		anti clockwise								anti clockwise							
SAE flange		2.3&4								2.3&4							
Injection		direct								direct							
Speed	rpm	1.500	1.800	2.000	2.150	2.300	2.500	2.650	2.800	1.500	1.800	2.000	2.150	2.300	2.500	2.650	2[illegible]
Mean piston speed	m/sec.	6.0	7.2	8.0	8.6	9.2	10.0	10.6	11.2	6.0	7.2	8.0	8.6	9.2	10.0	10.6	[illegible]
	(ft./min.)	(1.181)	(1.417)	(1.575)	(1.693)	(1.810)	(1.969)	(2.087)	(2.205)	(1.181)	(1.417)	(1.575)	(1.693)	(1.810)	(1.969)	(2.087)	(2.[illegible]
Continuous rating (A to DIN 6270) (10% overload)	BHP	41	49	53	56	58	–	–	-	51	61	66.5	70.5	73	–	–	
Mean effective pressure	kg/cm²	6.53	6.49	6.32	6.22	6.02				6.49	6.41	6.35	6.26	6.19	-	.	
	(lbs./sq. in.)	(92.88)	(92.31)	(89.89)	(88.47)	(85.62)				(92.31)	(91.17)	(90.32)	(89.04)	(88.04)	-	.	
Intermittent rating (B to DIN 6270) a) (heavy duty)	BHP	43	51	57	59	62	66	–	-	54	64	71	74	78	81	–	
b) (normal duty)	BHP	46	56	60	63	67	70	71	73	57	69	75	78	83	87	88	
Automotive rating to DIN 70020	BHP	–	–	–	–	67	70	71	73	–	–	–	–	83	87	88	
Mean effective pressure	kg/cm²					6.95	6.69	6.38	6.72					6.89	6.64	6.34	[illegible]
	(lbs./sq. in.)					(98.85)	(95.15)	(90.74)	(88.47)					(97.99)	(94.44)	(90.18)	[illegible]

		F4L 912	F5L 912
Maximum torque referred to automotive rating to DIN 70020	mkg (ft. lbs.)	22.4(162.0)	27.8(201.0)
	at rpm	1.500	1.500
Minimum speed during continuous operation	rpm	1.500	1.500
Minimum speed during idling	rpm	650 – 700	650 – 700
Bore / stroke	mm (in.)	100/120(3¹⁵⁄₁₆/4²³⁄₃₂)	100/120(3¹⁵⁄₁₆/4²³⁄₃₂)
Displacement	ltrs. (cu. in.)	3.77(230.1)	4.71(287.4)
Compression ratio		17	17
Fuel consumption at continuous rating A and 1,800 rpm			
full load	g/HPhr (lbs./HPhr)	158(0.354)	158(0.354)
¾ load	g/HPhr (lbs./HPhr)	163(0.365)	166(0.372)
½ load	g/HPhr (lbs./HPhr)	180(0.403)	185(0.414)
Lube oil consumption	kg/hr (lbs./hr)	0.1(0.220)	0.11(0.243)
Starting method *		EL.DL.F	EL.DL.F
Permissible inclination permanent/temporary			
fore or aft	deg	30°/40°	32°/37°
to either side	deg	20°/30°	20°/30°
Injection pump		Bosch	Bosch
Charging Generator		Bosch	Bosch
Net weight			
manually started engine	kg (lbs.)		-
electr started engine	kg (lbs.)		
with sheet metal oil sump	kg (lbs.)	320(705)	
with cast oil sump	kg (lbs.)	330(727)	405(893)
Shipping space, seaworthy packed	m³ (cu. ft.)	1.2(42.4)	1.2(42.4)
Seaworthy packing % of net weight	%	25	25

* H = manual EL = electric DL = compressed air F = inertia

tions

2.500	2.650	2.800
10.0	10.6	11.2
(1.980)	(2.087)	(2.205)
–	–	–
.		
.	.	
81	–	–
87	88	92
87	88	92
6.64	6.34	6.27
(94.44)	(90.18)	(89.18)

FGL 912							
6 anti clockwise 2.3&4 direct							
1.500	1.800	2.000	2.150	2.300	2.500	2.650	2.800
6.0	7.2	8.0	8.6	9.2	10.0	10.6	11.2
(1.181)	(1.417)	(1.575)	(1.693)	(1.810)	(1.989)	(2.087)	(2.205)
61	73	80	86	88	–	–	–
6.47	6.45	6.36	6.28	6.10	.	.	.
(92.02)	(91.74)	(90.46)	(89.32)	(86.76)	.		
86	77	86	89	94	98	–	–
80	23	90	94	100	104	108	110
–	–	–	–	100	104	108	110
				6.92	6.62	6.36	6.26
				(98.43)	(94.16)	(90.46)	(89.04)

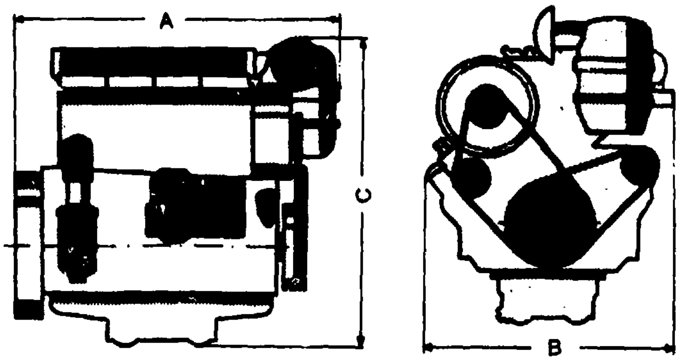
BFGL 913							
6 anti clockwise 2.3&4 direct							
1.500	1.800	2.000	2.150	2 300	2.500	2.650	2.800
6.25	7.5	8.3	8.95	9.6	10.4	11.05	11.7
(1.230)	(1.476)	(1.633)	(1.761)	(1 870)	(2.047)	(2.175)	(2.303)
96	112	123	130	136	–	–	–
9.3	9.14	9.03	8.88	8.62			
(132.2)	(130.0)	(128.4)	(126.3)	(122.6)			
100	119	130	136	142	148	–	–
106	125	137	144	152	160	160	160
–	–	–	–	152	160	160	160
				9.71	9.41	8.87	8.39
				(138.1)	(133.8)	(126.1)	(119.3)

FGL 912	BFGL 913
33.7(243.7)	50.0(361)
1.500	1.650
1.500	1.500
650 – 700	650 – 700
100/120(3[illegible]/4[illegible])	107/125(4[illegible]/4[illegible])
5.65(346)	6.17(373.5)
17	15 5
157(0.352)	157.0(0.352)
164(0.367)	161.5(0.361)
182(0.408)	174.0(0.389)
0 17(0.265)	0.14(0.309)
EL.DL.F	EL.DL.F
28°/34°	28°/34°
20°/30°	20°/30°
Bosch	Bosch
Bosch	Bosch
	–
	–
436(959)	485(1069)
445(981)	515(1135)
1.7(42.4)	1.3(45.9)
25	25

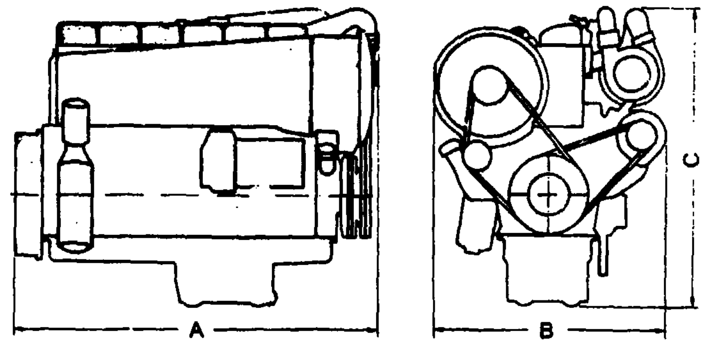
Arthur D Little, Inc.

2

F4L 912
through
F6L 912



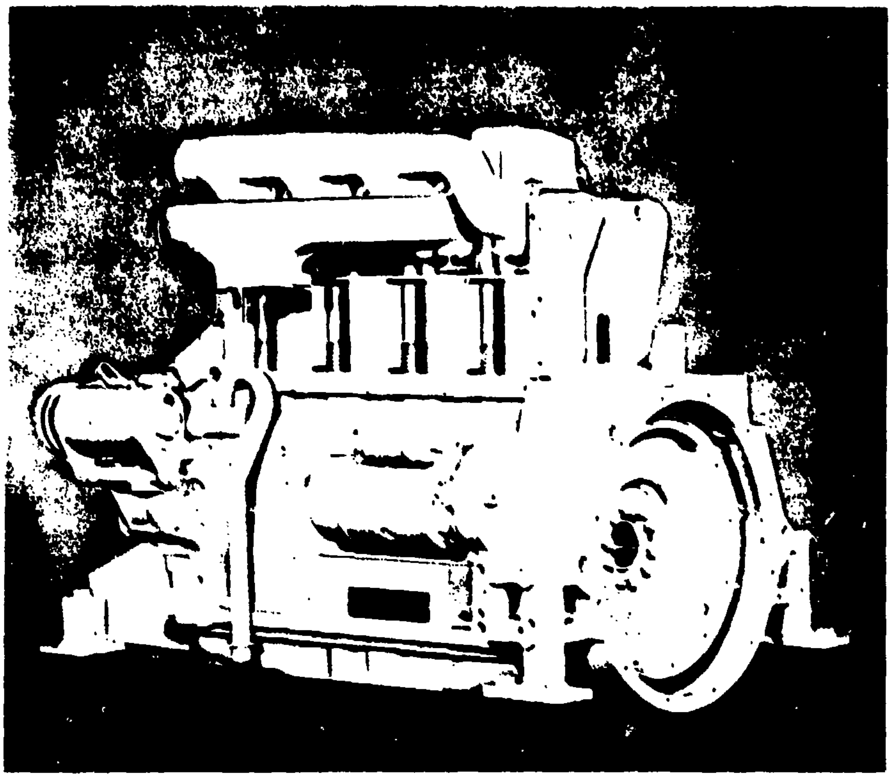
BF6L913



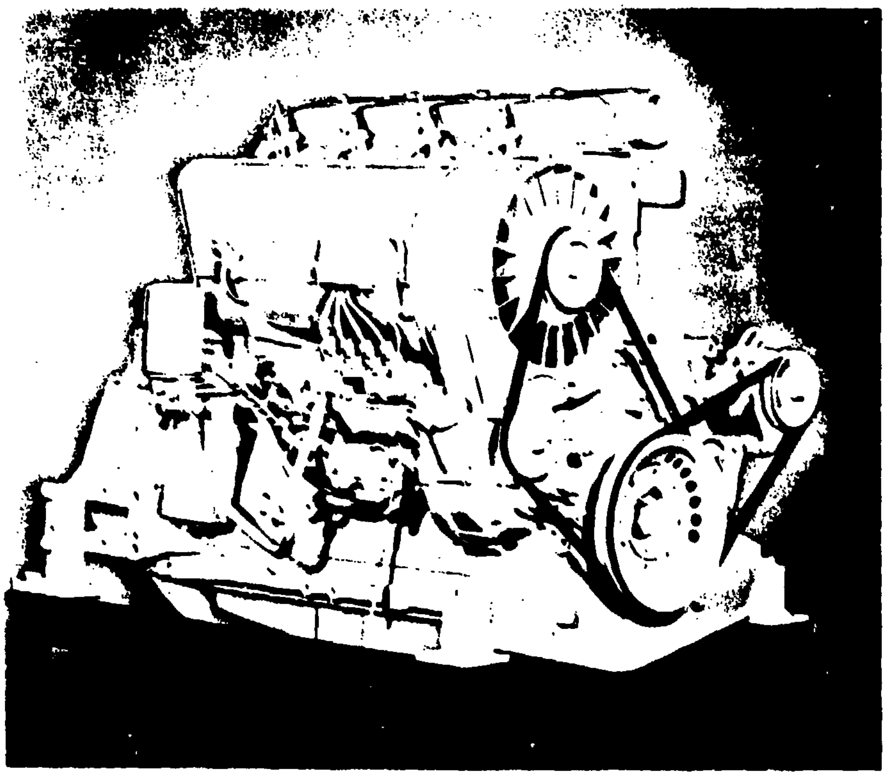
MODEL	A MM (Inches)	B MM (Inches)	C MM (Inches)
F4L 912	901 (35 15/32)	663 (25 5/16)	840 (33 1/16)
F5L 912	945 (37 13/64)	663 (25 5/16)	880 (34 41/64)
F6L 912	1165 (45 7/8)	663 (25 5/16)	855 (33 27/32)
BF6L913	1123 (44 1/4)	709 (27 7/8)	921 (36 1/4)

Figure 2-2

Dimensions of Deutz Diesel Engines

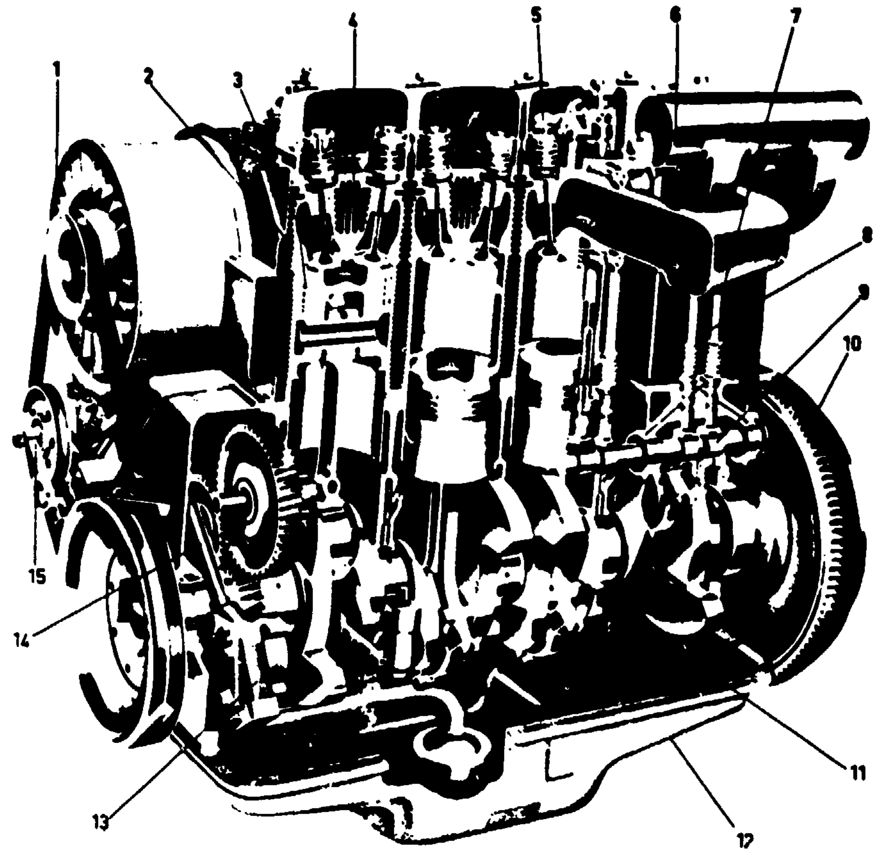


Exhaust Side



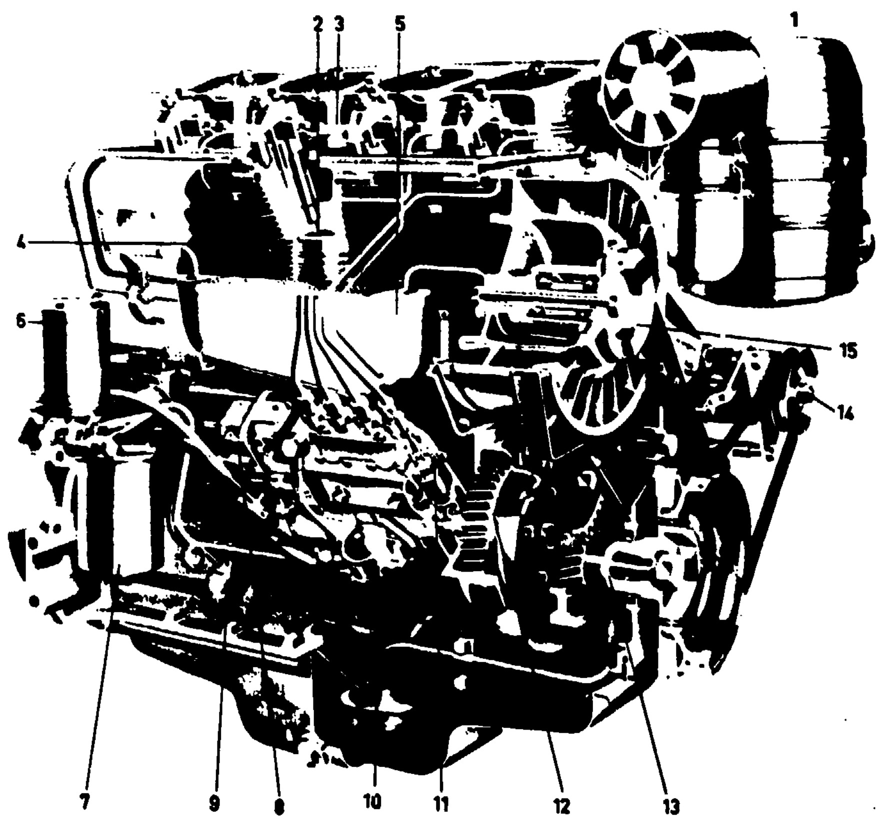
Service Side

Figure 2-3 Deutz Four-Cylinder Diesel, Model F4L 912



Exhaust Side

1 c ling fan
2 cooling fins
3 piston and (direct injection type) combustion chamber
4 light metal cylinder head and inlet/outlet valve assy.
5 rocker arm
6 induction manifold
7 exhaust manifold
8 pushrod and duct assy.
9 camshaft
10 flywheel and ring gear assy.
11 crankshaft and counterweight assy.
12 sump
13 lube oil pump
14 timing gear train
15 cooling fan



Service Side

1 oil bath type air cleaner and preliminary filter assy.
2 injectors
3 cylinder head cover
4 finned cylinder barrel
5 block-type oil cooler
6 fuel filter
7 lube oil filter
8 governor and speed control lever assy.
9 dipstick
10 fuel lift pump
11 fuel injection pump
12 timing gear train
13 lube oil pump
14 dynamo/alternator
15 cooling fan

Figure 2-4 Cutaway View of Deutz Four-Cylinder Diesel Engine

Table 2-3

Lister Diesel Specifications

ENGINE		HR2	HR3	HR4	HR6	HRS6
BS [illegible]9:1958 (bhp) rating	2200 rev/min	29.5	44.25	59	88.5	—
	2000 rev/min	27.5	41.25	55	82.5	102
	1800 rev/min	25.0	37.5	50	75.0	96
	1500 rev/min	21.5	32.25	43	64.5	80
	1200 rev/min	17.0	25.5	34	51.0	—
Din 'B' (PS) rating	2200 rev/min	32.9	49.3	65.8	98.7	—
	2000 rev/min	30.7	46	61.3	92	113.8
	1800 rev/min	27.9	41.8	55.8	83.6	107.1
Maximum gross b.h.p. at 2200 rev/min		36.75	55	73.5	111	—
Num[illegible] cylinders		2	3	4	6	6
Bore x [illegible]ke mm (in.)		107.95 x 114.3 (4¼ x 4½)				
Displacement—litres (in³) ...		2.09 (127.5)	3.135 (191.25)	4.18 (255)	6.27 (382.5)	6.27 (382.5)
b.m.e.p. 1500 rev/min—bar (lbf in²)			6.13 (88.9)			7.53 (109.2)
Fuel consumption at full load —g/bhp/hr (lb/bhp/hr)						
2200 rev/min		206 (0.45)	197 (0.43)	187 (0.41)	185 (0.40)	—
1800 rev/min		188 (0.41)	181 (0.40)	177 (0.39)	175 (0.38)	175 (0.38)
1500 rev/min		180 (0.40)	178 (0.39)	176 (0.39)	174 (0.38)	174 (0.38)
Lubricating oil consumption ...		Less than 0.75% of full load fuel consumption				
Weight of bare engine	kg.	280	370	432	560	626
	lb.	620	820	960	1245	1380

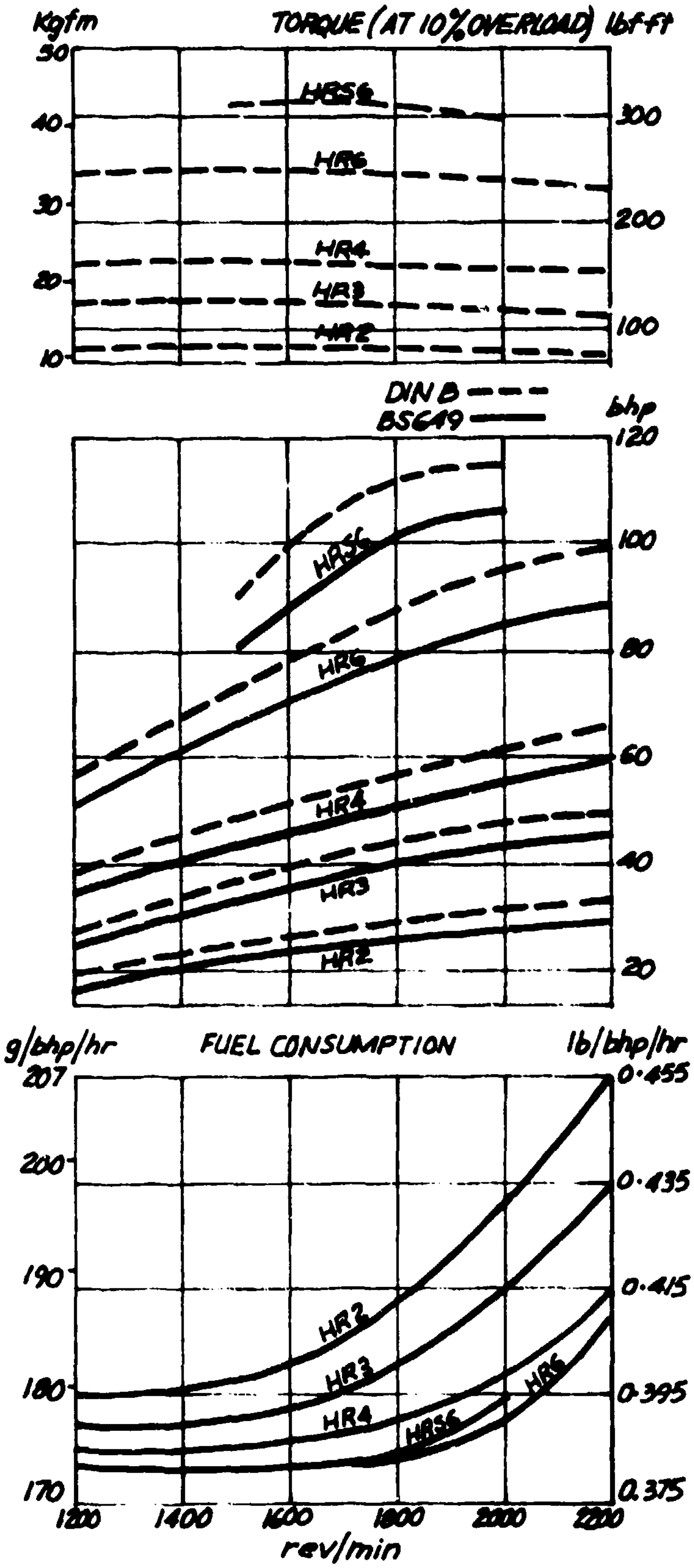


Figure 2-5

Power Curves for Lister Diesel Engines

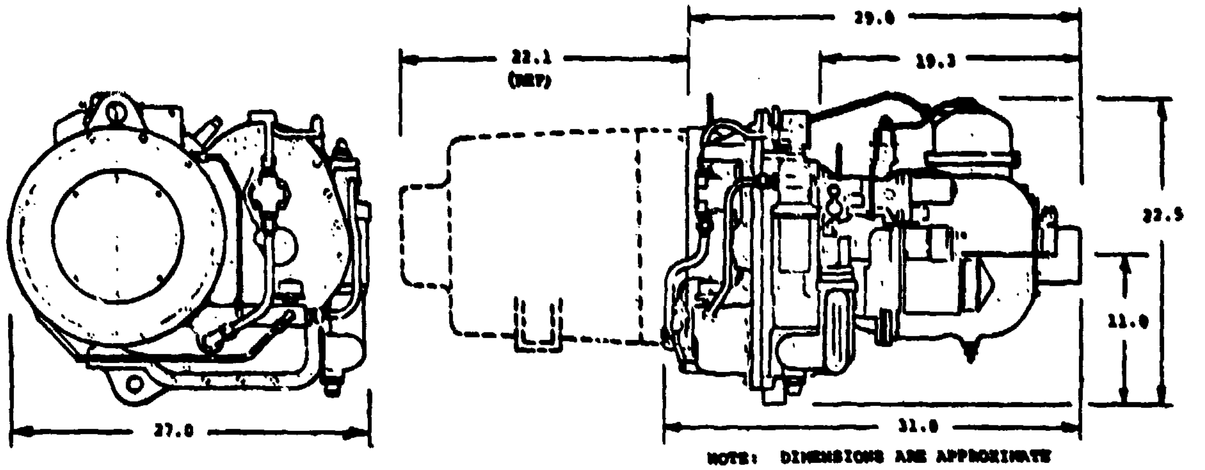
The appropriate Garrett model is the GTP36-51, a 75-hp turbine used mainly to drive the alternator in military portable generator sets. The principal specifications are shown in Figure 2-6, and a cross-section of the engine is shown in Figure 2-7. (The dotted line in Figure 2-5 represents an alternator and should be ignored for this application.) Output speeds ranging from 1500 to 12,000 rpm can be obtained with optional gearboxes. The nominal speed can be varied about ±10% by manipulating the fuel control spring.

The corresponding Turbomach machine is the T-62T-32 version of the Titan. The principal specifications and dimensions are given in Figure 2-8, and a cutaway view is shown in Figure 2-9. Because the output pad on the existing gearbox runs at 8000 rpm, an adaptor would be required to drop the speed to the 2500-rpm range of interest. This would add about 4 in. to the output pad, which would increase the overall length of the engine only slightly. Turbomach states that the speed can be varied from 70% to 100% of nominal through a throttling arrangement on the engine governor.

A much more detailed tradeoff analysis would be needed to select between these comparable units, once the actual application and requirements have been established. However, it can be seen that they are quite similar to one another and can be compared with the available diesels as suitable primary power sources.

As noted before, the major advantages of the gas turbine over the diesel are:

- its ability to be stored upside down without adverse effect or requiring preparation prior to subsequent use, and
- its low weight and small size for a given power level.



NOTE: DIMENSIONS ARE APPROXIMATE

ENVELOPE DATA

WEIGHT: (W/GEARBOX & ACCESSORIES) 370 LB. (DRY) INLET AREA: 50.0 SQ. IN.

EXHAUST AREA: 12.5 SQ. IN. (4 IN. DIA.)

PERFORMANCE DATA AND LEADING PARTICULARS:

FUELS: KEROSENE (JET FUELS) AND DIESEL FUELS. UNLEADED GASOLINE, LEADED GASOLINE (EMERGENCY)

OILS: NUMEROUS MINERAL AND SYNTHETIC OILS ARE APPROVED

DIRECTION OF ROTATION: CCW (FACING OUTPUT PAD)

SHAFT HP: (SEA LEVEL, 60°F DAY)
- *o CONTINUOUS DUTY: 75 HP
- **o STANDBY DUTY: 82 HP

SPEED: 80,000 RPM

*CONTINUOUS DUTY. THE RATING FOR LONG-LIFE ECONOMICAL PERFORMANCE WITH CONTINUOUS HEAVY DUTY LOADS.

**STANDBY DUTY. THE MAXIMUM HORSEPOWER OBTAINED WITH REDUCED LIFE, WITHOUT SIGNIFICANTLY DEGRADING ENGINE RELIABILITY.

STANDARD FEATURES:

- o FULL CONTAINMENT-COMPRESSOR AND TURBINE
- o SINGLE CAN - FIELD REPLACEABLE COMBUSTOR
- o IGNITION SYSTEM - LOW ENERGY
- o LUBE SYSTEM - EXCEPT HEAT EXCHANGER
- o FUEL SYSTEM - LOW PRESSURE
- o OVERSPEED AND OVERTEMP PROTECTIVE SENSORS (MONOPOLE AND THERMOCOUPLE)
- o ELECTRICAL STARTER - 24 VDC
- o OUTPUT PAD - SAE J617a-4
- o OUTPUT PAD SPEED - 3600/3000 RPM, BUILT-IN GEAR CHANGE
- o 3% SPEED ADJUSTMENT
- o OIL LEVEL DIPSTICK
- o CAST PARTS - TURBINE AND COMPRESSOR
  - TURBINE HOUSING
  - COMPRESSOR INLET AND SCROLL
  - GEARBOX

OPTIONAL FEATURES:

- o ELECTRONIC LOGIC PACKAGE (MATED TO ACCEPT STANDARD MALFUNCTION INDICATOR AND START/OPERATION/SPEED CONTROL FUNCTIONS)
- o GRAVITY FEED FUEL TANK
- o 50/60 HZ, 30 KW GENERATOR WITH REGULATOR AND BATTERY CHARGER
- o OUTPUT SPEEDS: 1500, 1800, 6000, 8000, 12,000 RPM
- o OUTPUT PAD - AND 20006-10 OR AND 20002-5
- o OUTPUT SHAFT

Figure 2-6

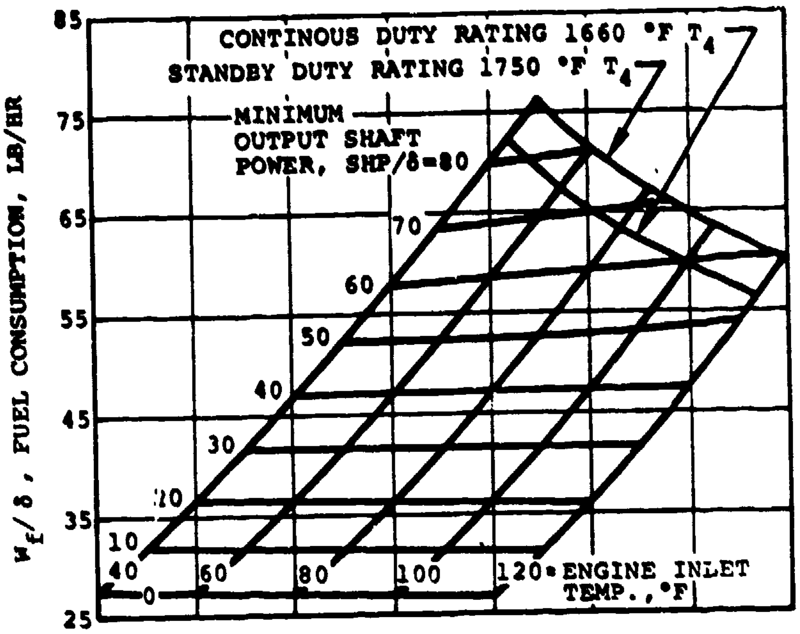
Preliminary Specifications for Garrett Turbine, Model GTP36-51

• PERFORMANCE SHOWN IS FOR SEA LEVEL OPERATION WITH NO INSTALLATION LOSSES

• ALTITUDE PERFORMANCE MAY BE ESTIMATED BY REDUCING HORSEPOWER SHOWN ON CURVE BY 3% FOR EVERY 1000 FT. GAIN IN ALTITUDE TO 10,000 FT.

NOTES:

1. FUEL LOWER HEATING VALUE EQUALS 18,400 BTU/LB
2. ENGINE INLET TOTAL PRESSURE EQUALS STATIC PRESSURE AT ENGINE EXHAUST EQUALS AMBIENT PRESSURE
3. OUTPUT SHAFT SPEED EQUALS 3600 RPM
4. DELTA EQUALS ENGINE INLET TOTAL PRESSURE, IN. HG ABS, DIVIDED BY 29.92



CUSTOMER INSTALLATION CONSIDERATIONS

FUEL REQUIREMENTS:	
SUPPLY PRESSURE - LIQUID -	GRAVITY TO 20 PSIG
TEMPERATURE -	-65° TO +140°F
ELECTRICAL REQUIREMENTS:	
STARTING -	24 V CONSISTING OF 2 SERIES 2 HN BATTERIES OR EQUIVALENT - 45 AMP-HR SUPPLY
OPERATION -	18-30V D.C.
OPERATING ENVIRONMENTS:	
TEMPERATURE - ENGINE INLET AIR -	-65°F TO +120°F
ALTITUDE -	SEA LEVEL TO 10,000 FT (30 TO 20 IN HG)
NOMINAL EXHAUST - GAS CHARACTERISTICS:	
FLOW -	0.92 LB/SEC (721 CFM)
TEMPERATURE -	1200°F
LUBRICATION REQUIREMENTS:	
CAPACITY -	5 QUARTS, HEAT EXCHANGER SIZED FOR 120 BTU/MIN @ 2.5 GPM

Figure 2-6 (Cont.)

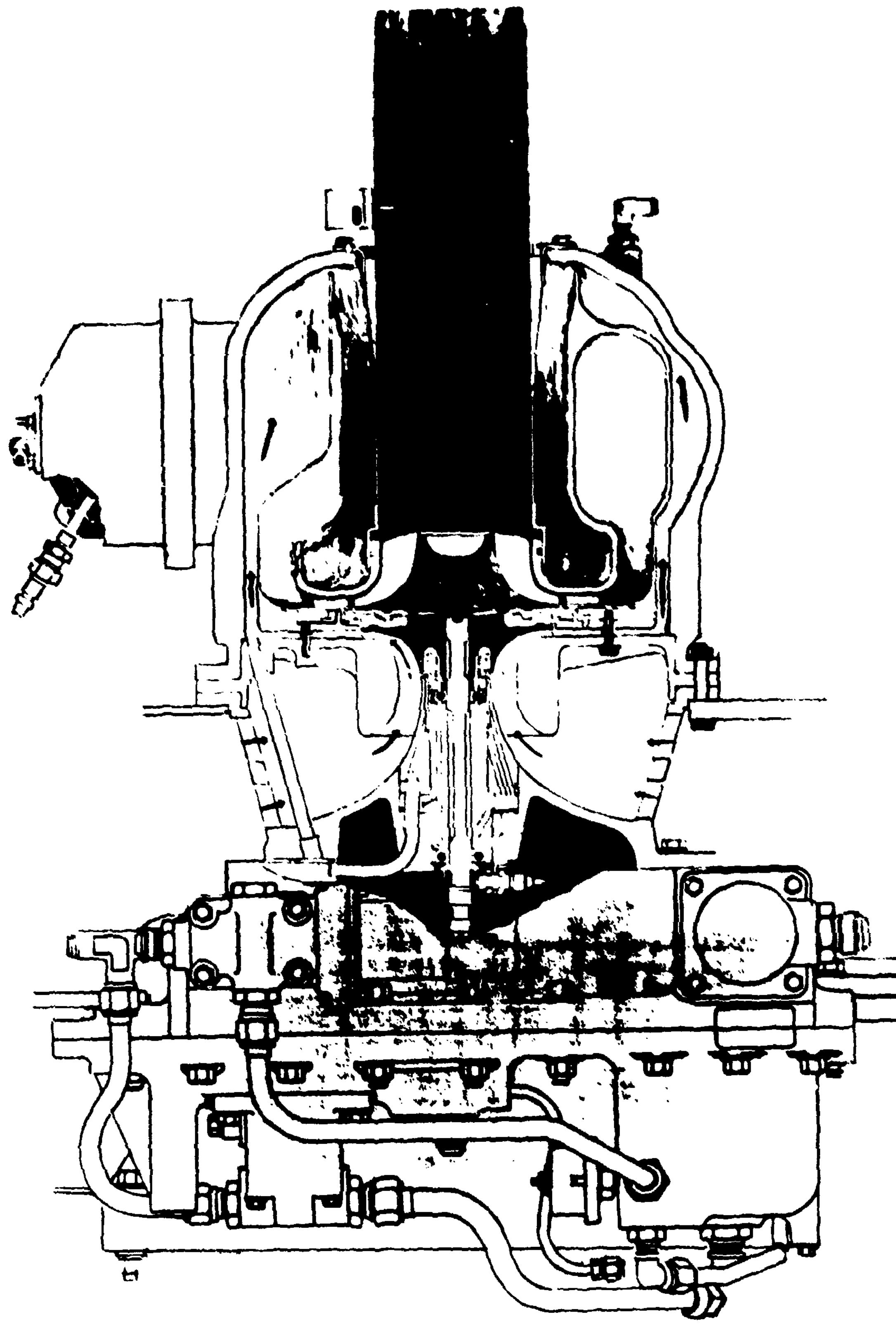
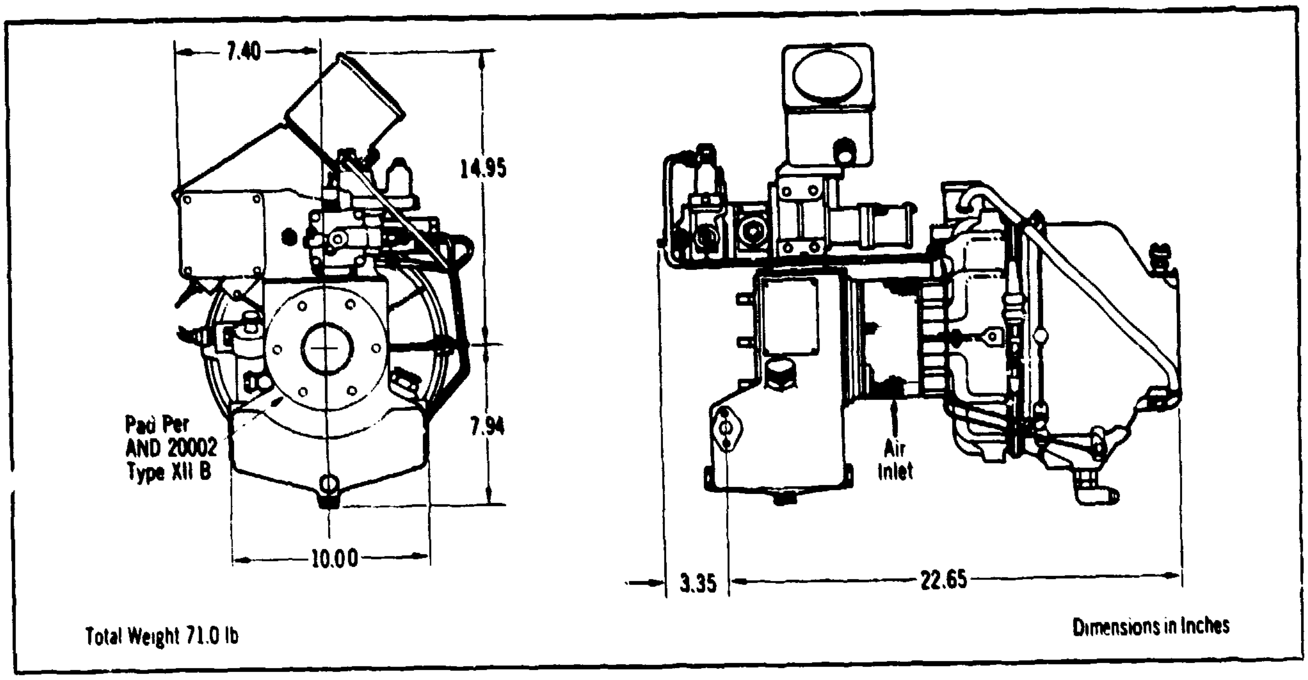


Figure 2-7

Cross Section of Garrett GTP 36-51 Gas Turbine Engine



Source: Solar Gas Turbines Div., International Harvester Co.

	T-62T-32A	T-62T-32
Rotational Speed (rpm)	72,226	61,091
Air Flow (lb/s)	1.45	2.14
Pressure Ratio	5.5	3.9
Compressor Efficiency (%)	78	76
Turbine Inlet Temperature (°F)	1360	1240
Turbine Efficiency (%)	86.5	84
Turbine Exhaust Temperature (°F)	820	840
Output Power (hp)	90.3	90.3
Output Pad Speed (rpm)	8000	8000
Fuel Flow (lb/hr)	68.3	103
Fuel Flow at Max. Power (lb/hr)	109	140
Max. Turbine Inlet Temp. (°F)	1840	1600
Max. Power (hp)	160	150
Best SFC (lb/hp-hr)	0.68	0.93
Overhaul Life (hr)	6000	1500

Figure 2-8

Turbomach Turbine Specifications

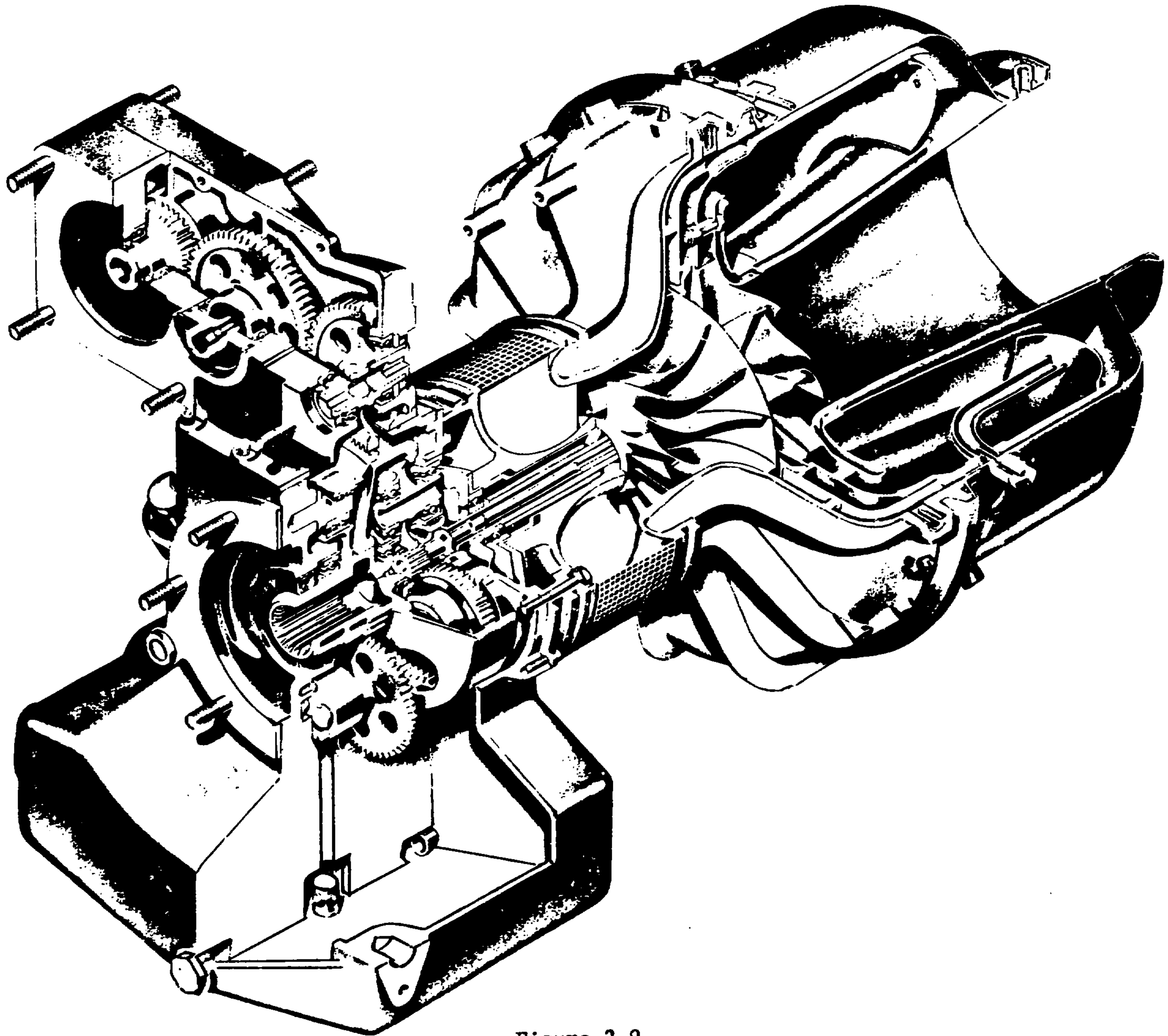


Figure 2-9

Turbomach Turbine -- Cutaway View

While these are worthwhile advantages, they carry with them the following penalties or disadvantages:

- limited speed control, making the use of a variable speed drive and/or a clutch likely;
- extremely hot exhaust gases;
- sophisticated maintenance procedures; and
- about twice the fuel consumption of a comparable diesel.

For these reasons, it may be concluded that the air-cooled diesel would be the preferred primary power source except in specific cases where the turbine must be used to permit inverted storage or to fit in a restricted space. In these cases, it can be expected that, unlike the diesel, an experimental development program will be required to adapt the gas turbine to the bridge drive system.

2.4 THRUSTERS

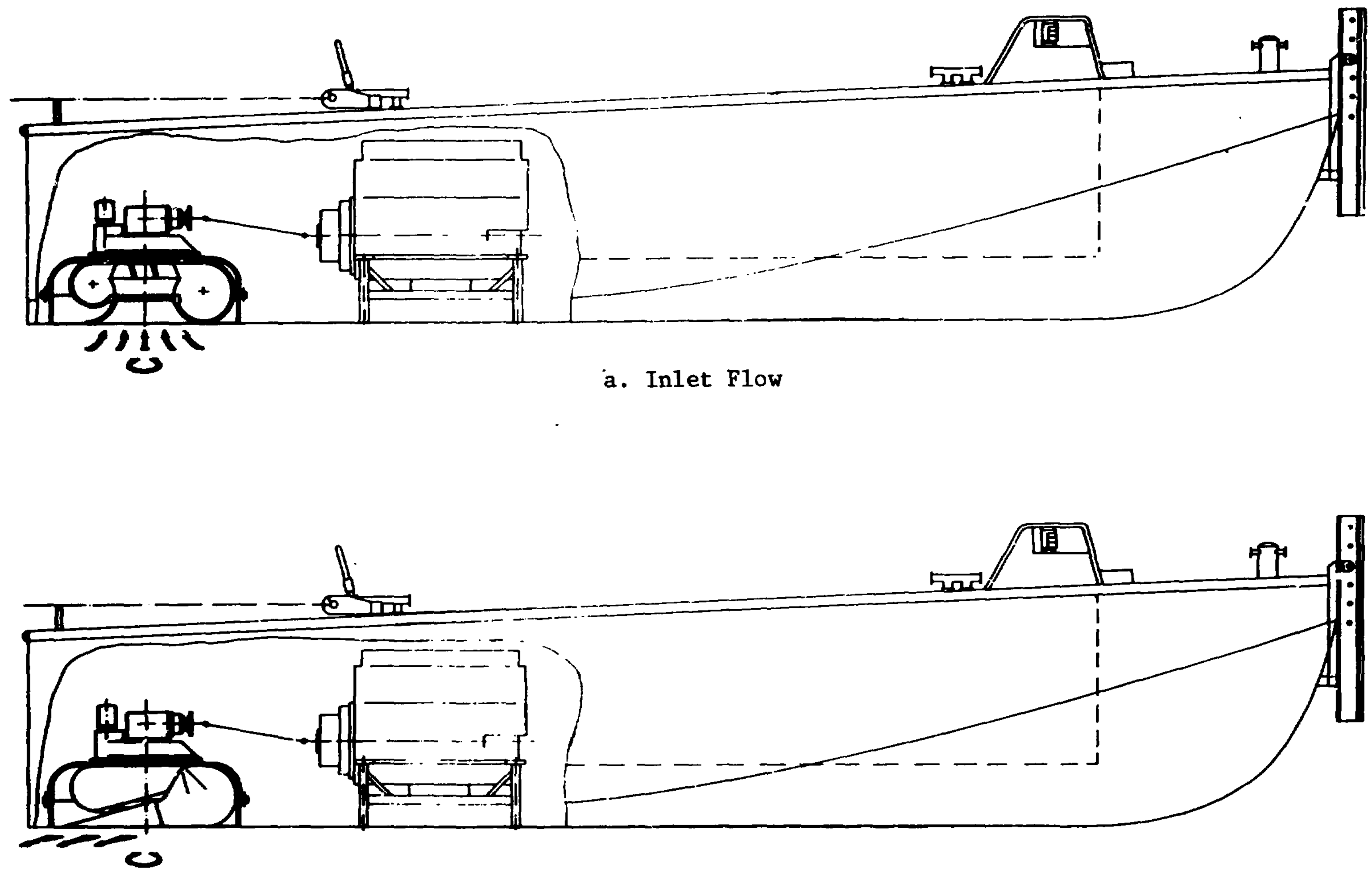
2.4.1 General Discussion

As noted earlier, the requirement that the bridge module operate in shallow water and the limitations imposed by the joining of individual modules makes it necessary for the thrust or propulsion unit to be either flush with the bottom of the module or installed at the bow and stern below the waterline.

Since the Schottel Pump-Jet is the only thruster we found that can be mounted flush with the bottom of the module, it was designated as the baseline thruster for the proposed system. Other possibilities include the Dowty Jet Thruster and various outboard-type propeller drives.

2.4.2 Schottel Pump-Jet

The Schottel Pump-Jet is basically a special mixed-flow pump designed to mount flush with the bottom of a boat, as shown in Figure 2-10. A



a. Inlet Flow

b. Discharge Flow

Figure 2-10

Schottel Pump-Jet Mounted in a Boat

cutaway drawing of the Pump-Jet is shown in Figure 2-11. Water enters axially (vertically upward), its energy is increased, and it is discharged tangentially with a downward component. The tangential discharge direction can be rotated 360° about the vertical axis by rotating the inner pump casing, giving complete directional control.

The standard unit is the SPJ-50, which requires a 1200-mm (47.2 in.) diameter opening in the hull bottom. The major specifications for this unit are given in Figure 2-12. The maximum thrust of 10,000 Newtons (2248 lb) is considerably more than that required for a bridge module, especially if two units are provided for maximum control.

A smaller special unit, the SPJ-32, was developed by Schottel for a German Army bridge pontoon. This unit, described in Figure 2-13, requires a hull opening of 1100 mm (43.3 in.) diameter and produces a maximum thrust of 7000 Newtons or 1574 lb, still somewhat large for this application.

The Schottel-Werft factory in Spay/Rhein, Germany was contacted to determine if a smaller unit could be designed which would be capable of a maximum thrust of 1000 lb, consistent with the requirements of each thruster in a bridge module. The result was the proposed SPJ-20 unit (Figure 2-14), which would produce 966 lb (4300 N) thrust at 2800 rpm. The shaft power would be 67 hp (50 kW) at 2800 rpm. This compact unit would require a hull opening of only 860 mm (34 in.) and would have an overall height of 520 mm (20.5 in.).

The overall height of the developed SPJ-32 can also be reduced, by means of a right-angle steering drive, to 650 mm (25.6 in.). As shown in Figure 2-15, this unit is capable of 900 lb thrust at a shaft speed of 2100 rpm and a shaft power of 60 hp.

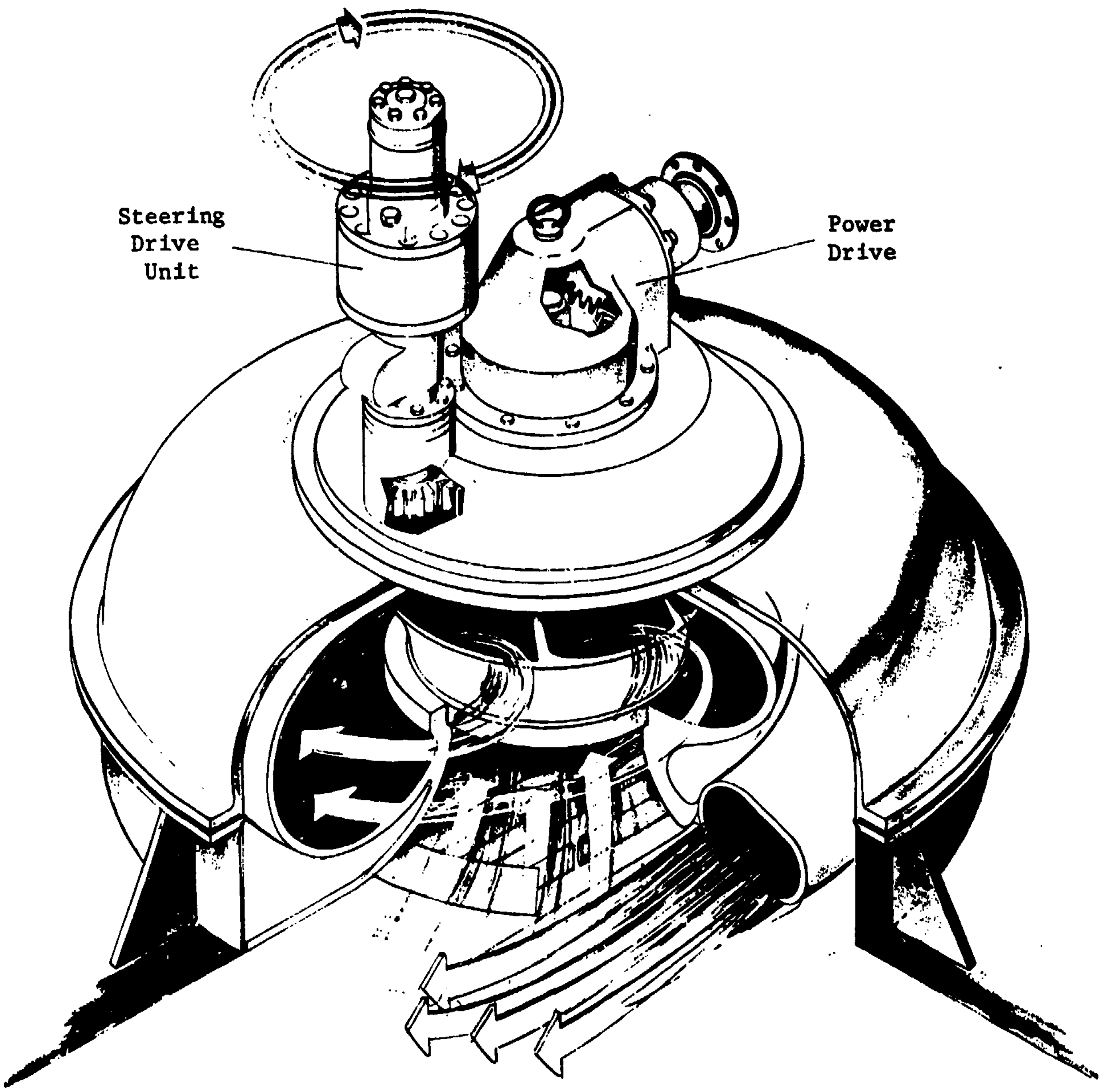
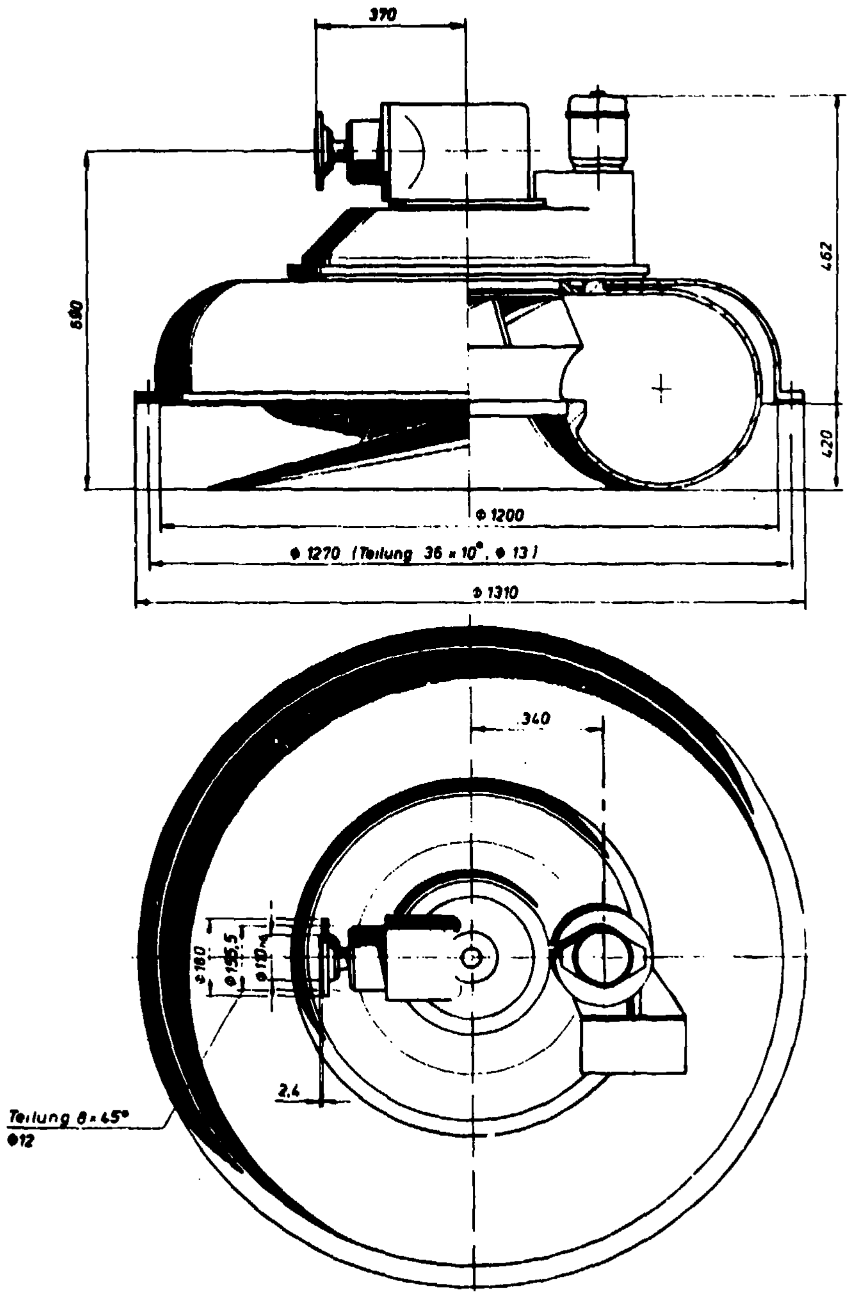


Figure 2-11

Cutaway View of Schottel Pump-Jet

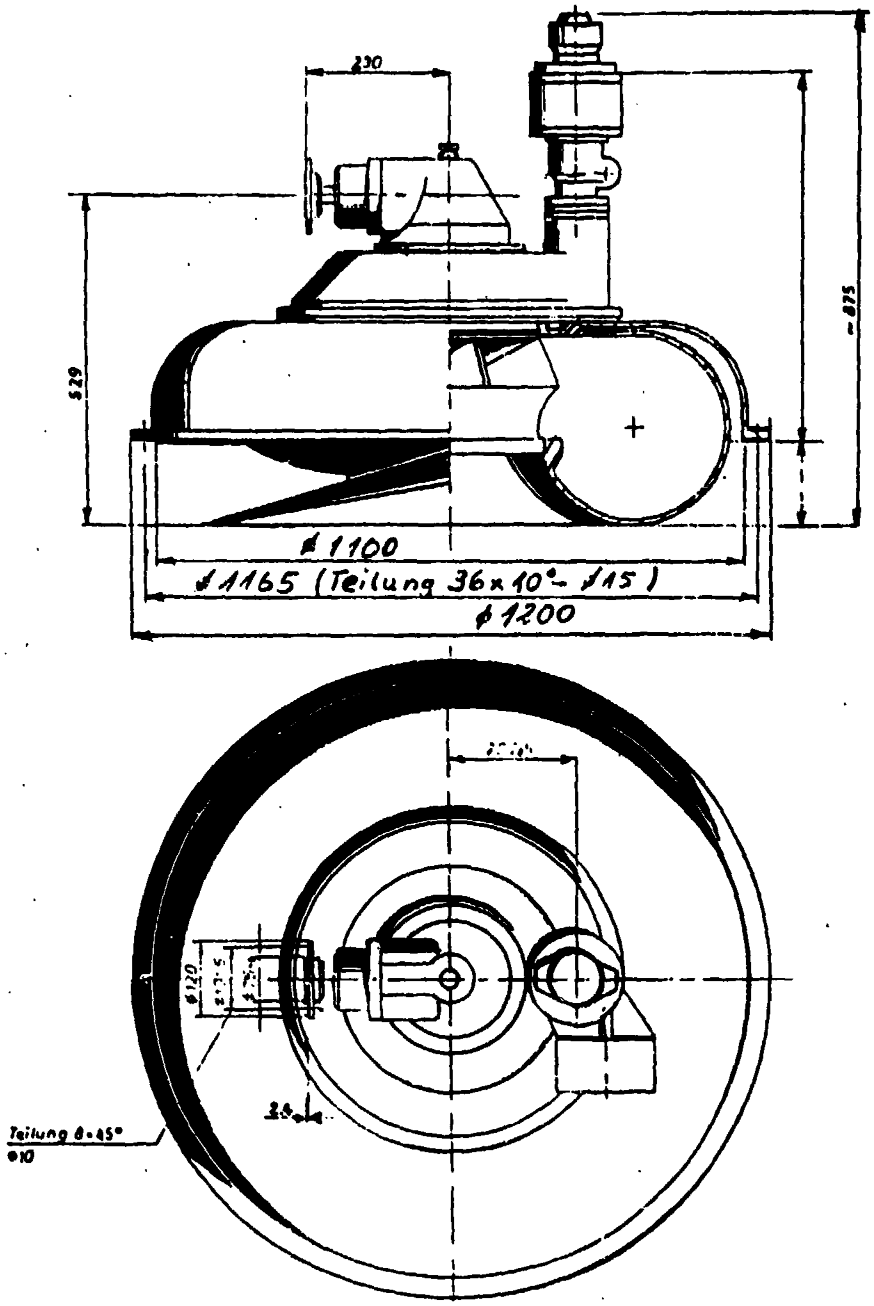


(All dimensions in millimeters)

Input Torque (Nm)	500
Input Speed (rpm)	2300
Loss of Displacement (kg)	~200
Thrust (N)	~10,000
Weight (kg)	~250

Figure 2-12

Schottel SPJ-50 Specifications

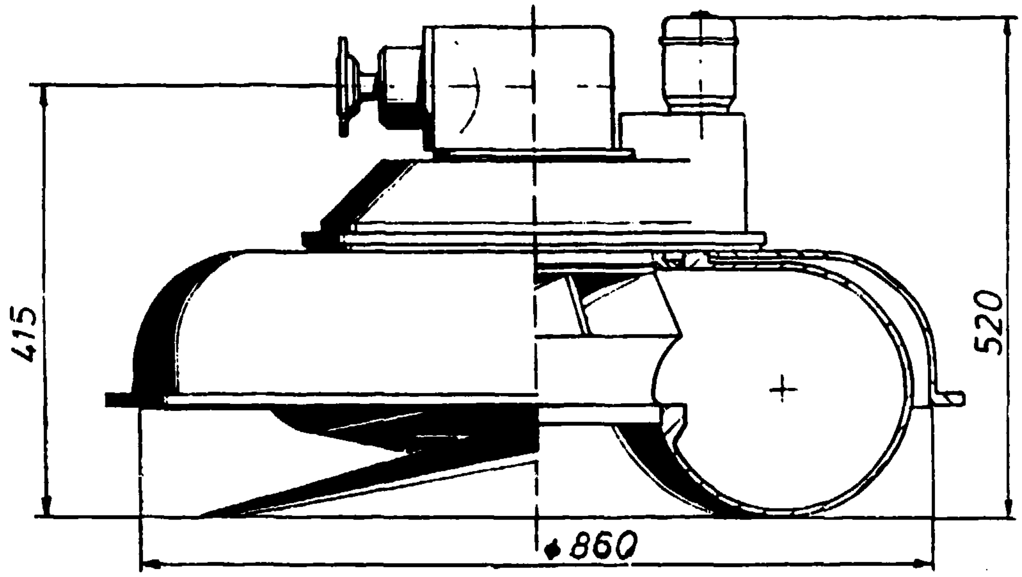


(All dimensions in millimeters)

Input Torque (Nm)	320
Input Speed (rpm)	2800
Displacement (kg)	~ 80
Thrust (N)	~ 7000
Weight (kg)	~ 110

Figure 2-13

Schottel SPJ-32 Specifications

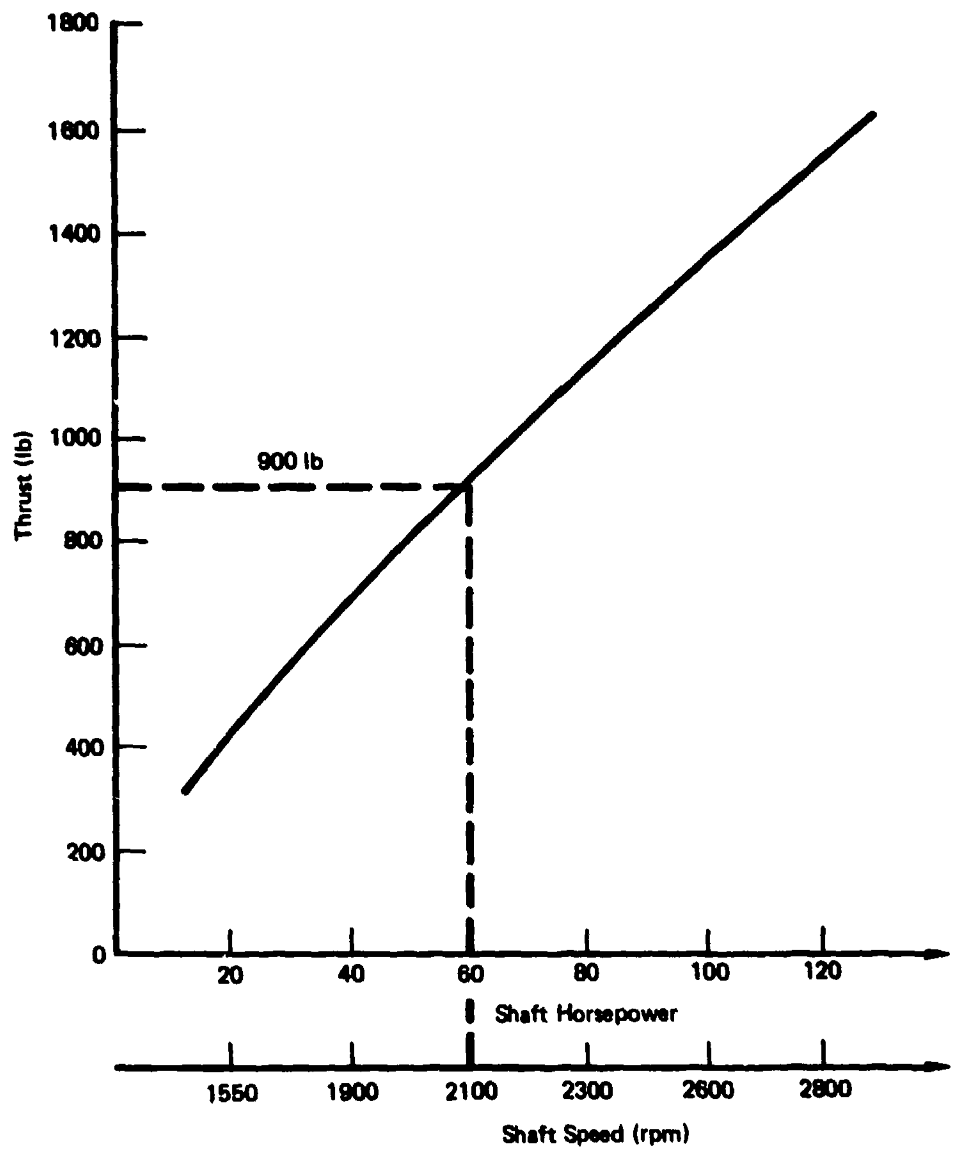


(All dimensions in millimeters)

Input Power (kW)	50
Input Torque (Nm)	174
Input Speed (rpm)	2800
[illegible]t (N)	4300
Weight (kg)	90 (Al)
	190 (GGG)

<u>Figure 2-14</u>

<u>Schottel SPJ-20 Specifications</u>



Source: Schottel-Werfte

Figure 2-15

Schottel SPJ-32 Thrust vs Shaft Speed and Power

Thus, either the developed SPJ-32 or the proposed SPJ-20 can be used for the bridge module. While the latter would be more compact, it would require higher shaft speed and input power to develop the necessary thrust.

2.4.3 Dowty Jet Thruster

A jet pump thruster with an axial configuration is manufactured by Dowty Hydraulic Units Ltd., Gloucestershire, United Kingdom. These units are produced in 300-mm (12-in.) and 400-mm (16-in.) impeller diameters, and each is available with one or two axial pump stages. The size of interest for this application is the 300-mm unit (Figure 2-16). Performance curves for the single-stage unit are shown in Figure 2-17.

The Dowty Hydrojet combines an axial flow pump with a steerable nozzle. The latter provides side-to-side steering over a 100° included angle. Astern or neutral thrust is obtained by changing the position of a scoop or hood over the nozzle.

The overall length of the single-stage unit inside the transom is approximately 40 inches, and the height required inside the hull is about 16 inches.

Because the primary thrust of this unit is axial, it would have to be located in the bow and stern sections of a bridge module. Thus, it is not as convenient as the Schottel unit for the three-piece module. In addition, it does not provide as flexible steering, being limited to ±50° about the nozzle centerline, or a total of 200° considering both ahead and astern thrust.

However, for the Ribbon Bridge, where all sections of the module are stored in positions 90° from the operating position and a gas turbine acquires some advantages, the Dowty unit has a potentially unique feature:

All dimensions in millimeters
with inch equivalents in brackets

Unit Type		Dimension X mm	Dimension X in	Weight kg	Weight lb
Single-stage	300/40	337	13·3	96	210
	300/60	337	13·3	96	210
*Two-stage	300/40-40	533	21	136	300
	300/40-60	533	21	136	300
	300/60-60	533	21	136	300

*as shown

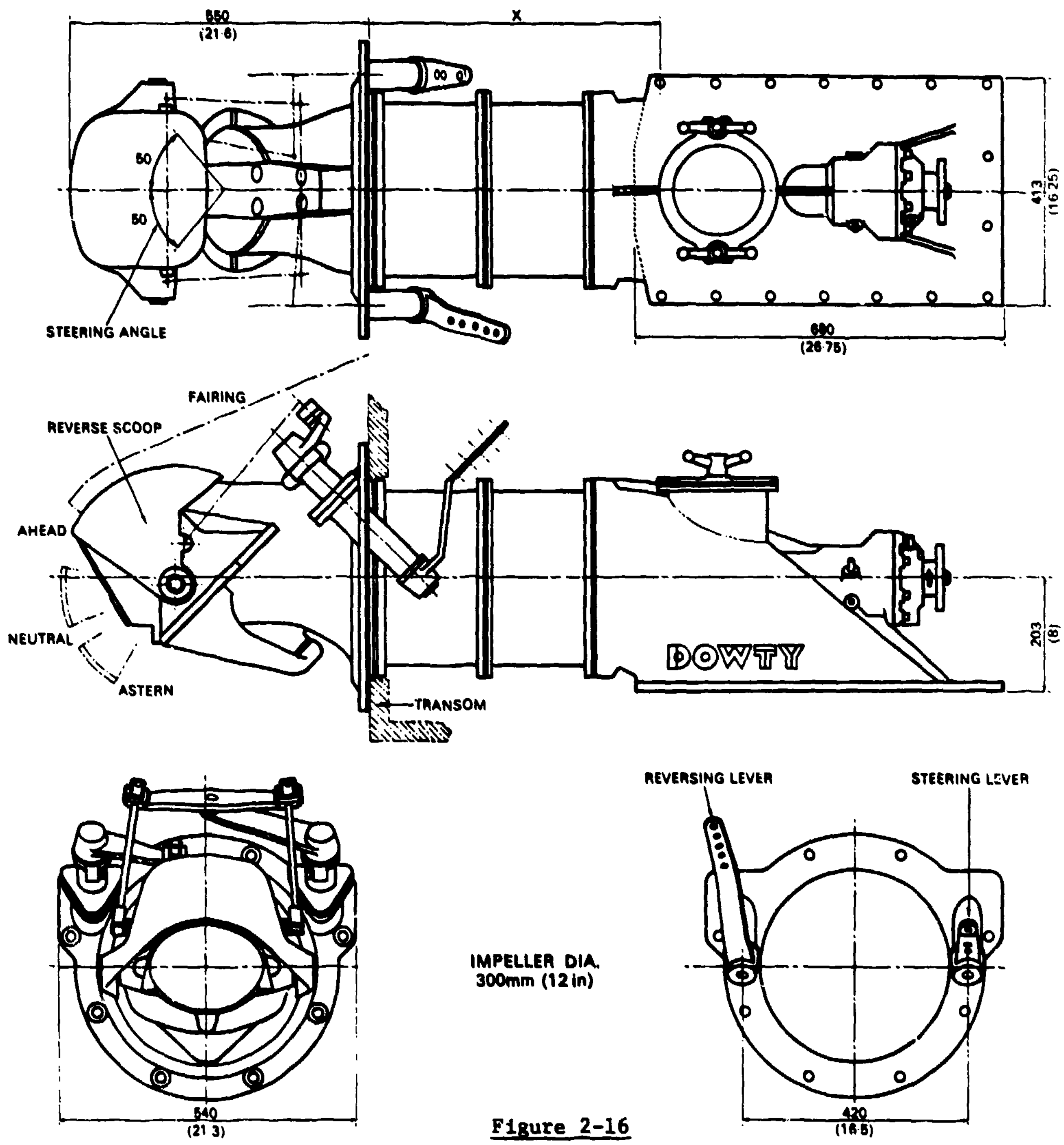
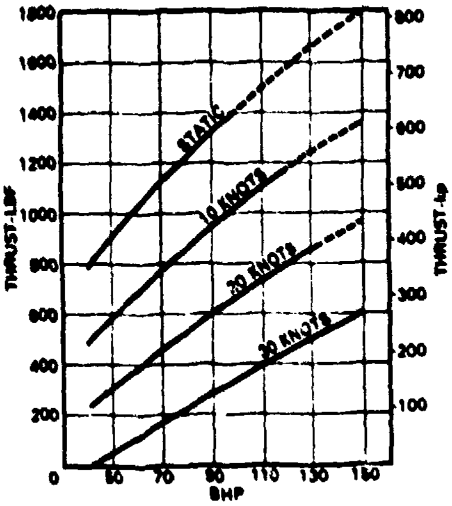
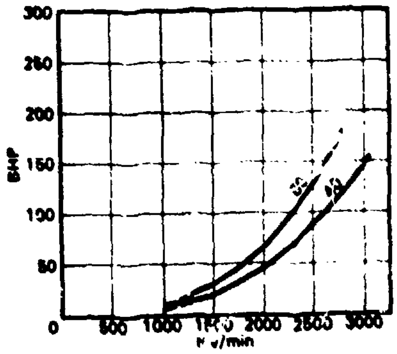


Figure 2-16

Dowty Hydrojet 300



Variation of Thrust
with Forward Speed



Power Absorption

40-Fine Pitch Impeller
60-Coarse Pitch Impeller

Source: Dowty Hydraulic Units Ltd.

Figure 2-17

Typical Performance Curves for Dowty Single-Stage Hydrojets the reversing scoop can be used to modulate the nozzle thrust. The scoop has a neutral position where no thrust is provided and, presumably, can be used to regulate speed. This is an important advantage when used with a gas turbine, which provides limited variation of shaft speed. Thus, the Dowty unit should be considered as a candidate for use with a gas turbine primary power source.

2.4.4 Outboard Propeller Drives

A number of outboard propeller drives are available on the market. Most of the standard units are designed for powerful engines and are configured to mount on the deck of a barge, with the propeller extending below the waterline and the engine on the deck.

Figure 2-18 shows a Schottel SRP-12 thruster connected to a diesel engine. (A similar unit is made by Murray & Tregurtha of Quincy, Mass., among others.)

Technical specifications for the Schottel SRP-12 unit driven by four different engines are listed in Table 2-4. The thrust obtainable with a 500-mm (19.7-in.) diameter propeller is 26 lb/hp; thus, a properly matched 35-hp engine could produce the desired 900-lb thrust.

For use in a bridge module, the prime mover must be located inside the hull with the drive shaft extending through the hull at the bow and stern. For proper adapting to a bridge module, a special outboard package would undoubtedly be required, but this could probably be based upon one of the standard packages, such as those listed in Table 2-4, with comparable thrust performance.

For a three-ponton bridge module, the only advantage of an outboard drive unit would appear to be its higher thrust efficiency (26 lb/hp versus about 13 lb/hp for the Schottel Pump-Jet). This would, of course,

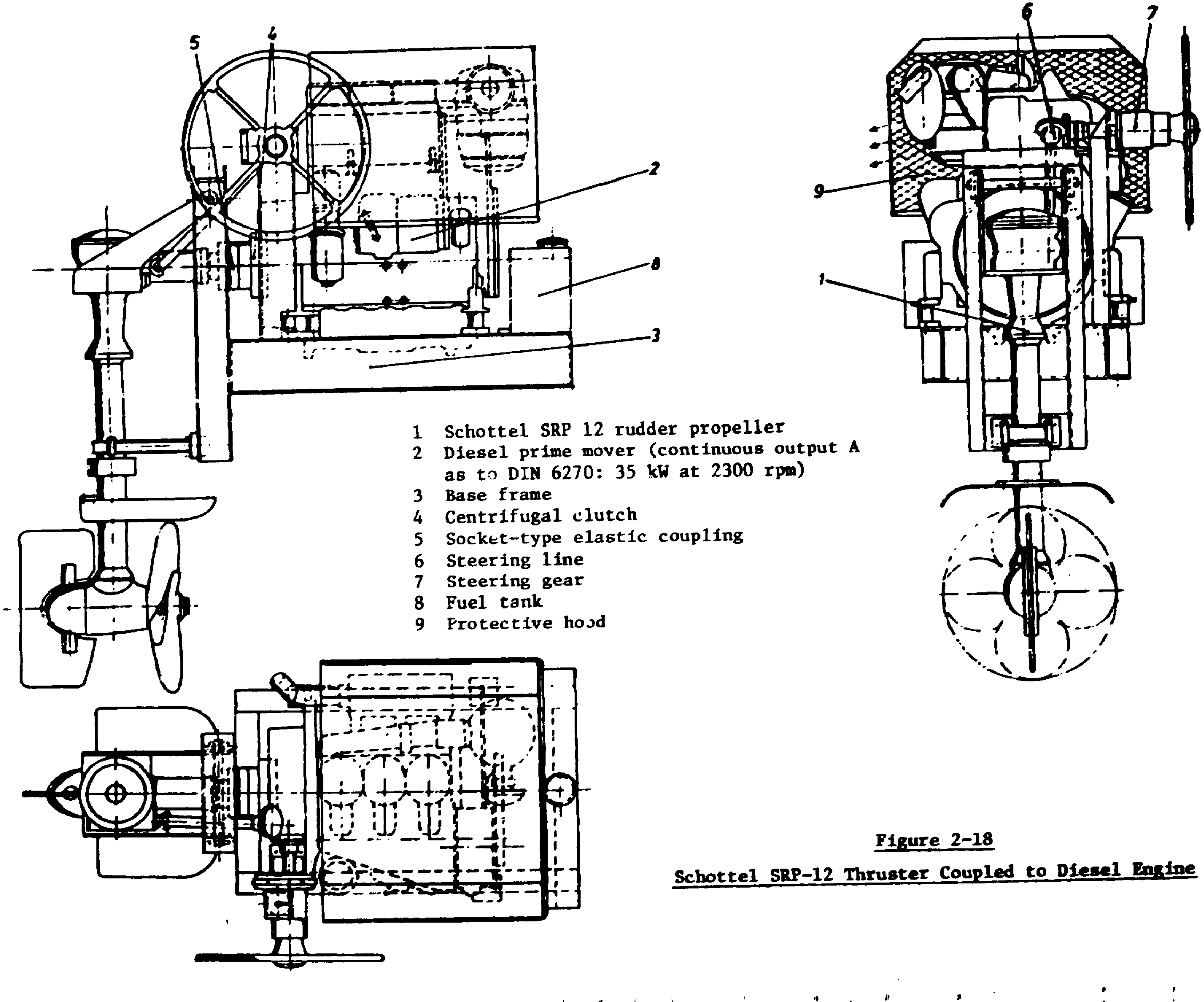


Figure 2-18

Schottel SRP-12 Thruster Coupled to Diesel Engine

Table 2-4

Schottel Propeller Thruster Specifications

	Item Designation	Prime Movers: VW industrial-type engine four-stroke controlled ignition engine	KHD four-stroke diesel engine	Hatz four-stroke diesel engine	MWM four-stroke diesel engine
Schottel Rudder Propeller	Type	SCHOTTEL rudder propeller SRP 12			
	Design	Tractor propeller			
	Propeller stem lenght	1000 mm 1)			
	Propeller material	Special cast aluminium			
	Propeller direction of rotation when viewed in the line of advance	Counterclockwise	Clockwise		
	Propeller number of revolutions at max. engine speed	1186 rpm	1190 rpm		
	Propeller diameter	500 mm			
Prime Mover	Type	126 A	F 3 L 912	D 108	D 327-3
	Type of design	Opposed cylinder engine	In-line engine		
	Propelling power as to DIN 70 020	34 KW	--	--	--
	Propelling power as to DIN 6270	--	35 KW	32 KW	
	Max. engine speed	3600 rpm	2300 rpm		
	Standard operating conditions in respect of engine power	760 mm Hg - 20 °C of intake temperature	736 mm Hg 20 °C of intake temperature		
		60 % relative humidity			
	Direction of rotation when viewed at flywheel	Counterclockwise			
	Displacement	1584 cm^3	2830 cm^3	3021 cm^3	2827 cm^3
	Bore / stroke	85,5/69 mm	100/120 mm	108/110 mm	100/120 mm
	Compression ratio	7,7:1	17:1		
	Type of cooling	Air cooled			
	Lubrication	Forced feed lubrication			
	Fuel	Gasoline of a RON of 90 min.	Gas oil H.C.V. = 10000 kcal/kg		
	Fuel consumption	16 lit/h 2)	Abt. 170 g/h.p.-h	Abt. 180 g/h.p.-h 3)	Abt. 175 g/h.p.-h
	Lubrication oil consumption	Abt. 1 g/h.p.-h	Abt. 2 g/h.p.-h	2 to 3 g/h.p.-h	1 to 2 g/h.p.-h
	Max. permissible inclination in service	Long. 15° Transv. 15°	Long. 20° Transv. 37°	Long. 10° Transv. 10°	Long. 25° Transv. 35°
	Starting by means of	Handcrank	Electric starter		
	Fuel tank	Abt. 22 liters	Abt. 30 liters		
Speed Reducer and Shifting Clutch	Speed reducer brand	Hermes type 113-01V			
	Gear ratio	1.57			
	Direction of rotation	Reversal			
	Shifting clutch	Fichtel & Sachs brand			
	Centrifugal clutch		Suco brand		
Weight	SAV basic version without SRP nor oil	Abt. 210 kg	Abt. 570 kg	Abt. 580 kg	Abt. 600 kg
	Rudder propeller without oil in case of PAL of 1000 mm	Abt. 60 kg			

1) Dimension from center of power inlet to center of proppeler shaft.
2) Measured at a continoue uniform load during one full hour.

allow a smaller engine. The penalties for this efficiency are:

- mandatory location in the bow and stern sections of the module;
- inconvenience in operation in shallow water; and
- vulnerability to damage from floating objects.

For these reasons, the outboard is not recommended as a thruster for the bridge module in any case where a jet thruster and matched primary power unit can be used.

2.5 THRUSTER-ENGINE MATCHING

2.5.1 Schottel Pump-Jet

From Figure 2-15 it can be seen that the Schottel SPJ-32 thruster absorbs 60 hp at the point where it is capable of producing 900 lb of thrust. The shaft speed at this operating point is 2100 rpm.

Referring back to Table 2-2, it can be seen that the four-cylinder Deutz diesel, Model F4L912, is capable of providing over 60 hp at 2100 rpm under intermittent conditions but that the five-cylinder engine, Model F5L912, would be required for continuous thrust at 900 lb. If 900 lb of thrust per unit or 1800 lb per bridge module is required for station-keeping in a strong current, continuous duty would be required.

The SPJ-32 can also operate at up to 2800 rpm. Comparing Figure 2-15 with Table 2-2 shows that intermittent operation up to 2600 rpm would be possible with the six-cylinder engine, Model F6L912. Figure 2-15 shows that this will provide a peak thrust, when needed, of about 1300 lb.

In summary, the optimum engine for the Schottel SPJ-32 is the Deutz F6L912 where there is sufficient room to mount it. If space is limited, the five-cylinder engine, Model F5L912, can be used to meet the nominal 900 lb/thruster requirement.

If the thrust requirement is reduced to 1300 lb/bridge module or 650 lb/thruster, the SPJ-32 can be operated at 1900 rpm, which requires only 40 hp. This can be supplied by the four-cylinder engine, Model F4L912.

The SPJ-20 Pump-Jet requires 57 hp at 2800 rpm to achieve a 900-lb thrust. However, as none of the Deutz diesels is rated for continuous duty at 2800 rpm, a gearbox with a small gear ratio would be required to obtain this thrust level. The F5L912 engine would likely be sufficient to supply the shaft power. Although no power absorption curve is available at this time, downrating the unit to 650 lb would probably allow operation at a shaft speed of about 2500 rpm, which would only require an F4L912 or smaller engine.

2.5.2 Dowty Jet Thruster

Optimum use of the Dowty thruster, as noted earlier, is obtained with a gas turbine drive. To achieve a static thrust of 900 lb per unit, the Model 300 requires about 50 hp, which it can absorb at about 1800 rpm.

Both of the gas turbines described in Section 2.3.3 are capable of providing shaft power well in excess of 50 hp. Proper matching would only require that the gas turbine (Garrett GTP36-51 or Turbomach T-62T-32) be obtained with a gearbox providing 1800-rpm shaft speed to the Dowty unit. The thrust would then have to be reduced by adjusting the reversing scoop on the thruster.

If the Dowty unit were driven by a Deutz diesel, the required continuous power at 1800 rpm places the operating point just above the capability of the four-cylinder Model F4L912. Thus, either a four- or five-cylinder engine could be used. Downgrading the thrust to 650 lb would appear to allow the use of the three-cylinder engine.

2.5.3 Outboard Propeller Drives

Again, optimum use of the outboard propeller drive for the bridge module is with a gas turbine drive. Taking the Schottel thrust characteristic of 26 lb/hp, a 900-lb thrust requires a shaft power level of 35 hp, which is well within the capability of the Garrett or Turbomach gas turbine with an appropriate gearbox. In fact, the much smaller Turbomach Gemini gas turbine might be suitable.

Since the gas turbine permits very little variation of speed and, thus, thrust, a variable-speed drive of some type would be required between it and the propeller thruster. Speed variation could be provided by a d-c generator and motor, a hydraulic slip-type drive, or a mechanical variable-speed drive; in the 35-hp range, however, all of these alternatives are large, heavy, and costly.

Another alternative is to use a thruster with a variable-pitch propeller. Although no standard controllable-pitch propellers are made in the size range of interest, the technology is available and a system could easily be obtained in the quantities of interest.

With a propeller drive, the power is reduced to the point where the three-cylinder diesel engine could probably be matched at maximum shaft speed and the four-cylinder would be ample. Reduction of the thrust requirement to 650 lb/unit would allow use of the three-cylinder engine.

3. RIBBON BRIDGE WITH INTEGRAL PROPULSION

3.1 INTRODUCTION

The Ribbon Bridge is a complete and mature system now in the Army inventory. Since original definition of the system, conditions and emphasis have changed; thus, some modifications and improvements of the bridge characteristics would be desirable. The current model of the bridge could become outmoded around 1993, but a major product improvement program could take half of the remaining 12 years to produce changes in the field. Developing a new Wet Support Bridge configuration or concept and carrying it forward through the several stages to deployment would, of course, take much longer and probably would coincide reasonably well with the expected obsolescence of the Ribbon Bridge; however, in this case, the Army would lack a bridge with improved capabilities for perhaps six years.

Potential changes and improvements to the Ribbon Bridge system must be viewed and evaluated in this context. Some of the alternatives suggested in this report may be too complex or costly in view of the overall expected life of the system, but others may be worth investigating. However, the current Ribbon Bridge is a complex system with many interrelationships between bridge components, transporters, and expected environmental constraints, both on land and in the water; a change at any point in the system can have far-reaching effects. We shall trace these effects as far as we can identify them and, if possible, quantify them.

3.2 DESCRIPTION OF PRESENT SYSTEM

The major component, the interior bay, consists of two roadway pontons and two bow pontons, hinged to fold into a "W" configuration. Each bay is 6.92 m (22 ft 8.5 in.) long and 8.15 m (26 ft 8 in.) wide in unfolded position.* When folded, it is 3.22 m (10 ft 6.6 in.) wide at the base plus projecting pads that bring it close to 3.5 m (11 ft 6 in.) in width. Height is 2.31 (7 ft 7 in.) when folded. Weight is 5443 kg (12,000 lb). These dimensions largely are governed by the roadability of the standard 5-ton Army chassis which is the basis of the transporter and by the constraints of the expected area of transport. The maximum interior height of a roadway ponton of an interior bay is 0.69 m (27 in.), which is reduced in many places by structural members. The principal tension member runs along the bottom of this ponton along its centerline and may not be interrupted.

The 8.2 m (27-ft) Bridge Erection Boat is also carried on a 5-ton truck and weighs 3090 kg (6800 lb) with cradle. It is powered by two 90-hp engines, draws slightly over one meter of water, and is propeller-driven. It is allocated to bridge companies at a ratio of about one boat per three bridge bays. It is designed to develop 3500 lb of thrust (1590 kg).

3.3 POTENTIAL AREAS OF IMPROVEMENT

Improvement of the Ribbon Bridge is desired in two principal areas:

(1) Logistics: A shift in tactical approach and a need to reduce the volume of the supply chain point toward integral propulsion of the bridge bays. This would give better rafting capabilities and would largely eliminate the need for erection boats.

(2) Water Velocity Limitations: Imposition of the requirement that the Ribbon Bridge be able to carry Military Load Class 70 further limits the water velocity in which it can be used; The two

*Note that "length" and "width" are measured in relation to the length and width of the assembled bridge; i.e., the "length" is measured in the direction traveled by vehicles on the bridge, and the "width" is the distance between the tips of the bow pontons, characteristics are mutually conflicting, and both are adversely affected by any weight increase associated with integral propulsion.

Various aspects of these two topics and possible alternative approaches are discussed in detail below.

3.4 INTEGRAL PROPULSION ALTERNATIVES

To be effective, a propeller-type drive must extend into the water flow. While in a bridge, it must not extend beneath the bottom, where it might be damaged at launch or low water. It cannot be at the sides where the bay is joined to adjacent units. Possibly it could be under the rake of the bows, but even then it would have to be retractable, as this surface forms part of the envelope of the bay in folded mode. The Pacific Car & Foundry study on use of two outboard engines modified to rotate into recesses into the bow pontons is an example of this approach. However, close examination of the restrictions imposed by the current Ribbon Bridge design indicates the incomparable advantages of the Schottel Pump-Jet type of propulsor: it is the only type of propulsion subsystem that does not project beyond the limits of the ponton and therefore does not require any retraction arrangement.

The roadway and walkway surfaces of the four pontons of the interior bay all mate when the bay is folded into its "W" configuration. The bottoms of the two central roadway pontons also mate. The undersides of the bows (the "rakes") and the vertical sides of all four pontons must be considered as immersed surfaces when the unit is afloat; except for the small freeboard, they are underwater and cannot be penetrated. Yet, with a short cycle of training/use/storage/maintenance, the prime mover inside the pontons must be accessible, both when the bay is folded and when it is afloat. Watertight bolted patch-plates are a possible solution but are not practicable because of the need for operational readiness on short notice. An access port in the middle of the roadway could be used when the bay is open, but a second access for use when the bay is stored would compromise the readiness and integrity of the unit.

A prime mover within a roadway ponton is limited in height to less than .69 m (27 in.). Most diesel engines in the required power range stand higher, but an air-cooled engine of suitable size might be found. The other major problem with diesel engines is that in the stowed position of the pontons they will be tipped 90° from their normal operating position. This is mainly a crankcase/oil sump problem and would probably require a design change. According to industry information, most engines can be tilted no more than about 20° during operation or (presumably) extended storage.

Gas turbines provide an alternative as prime movers and permit some savings in weight and space if they drive controllable-pitch propellers to avoid variable-speed gears. They about double the fuel consumption, however, and greatly increase the complexity of the drive and gearing. They can be stowed in any position, but they are more costly. Noise and exhaust problems are more severe than with the diesel, and the intake air must be cleaned more thoroughly.

The gasoline-fueled outboard installation suggested by Pacific Gas and Foundry is not considered further because of the fuel, because it appears to be underpowered, and because the retraction mechanism is vulnerable and the recesses increase the floodability--if not the initial buoyancy--of the pontons.

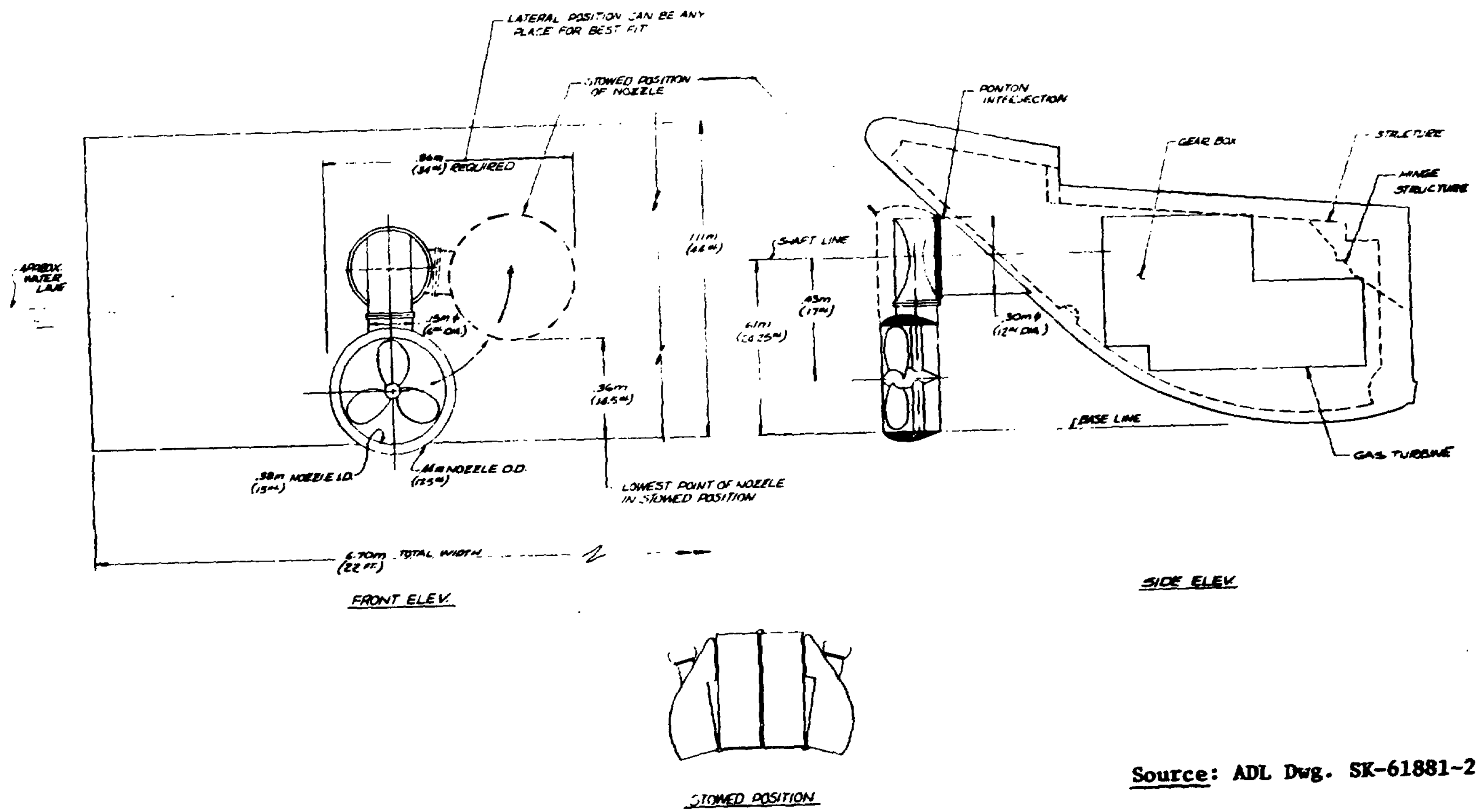
Although no Schottel Pump-Jet of the specified output is now in production, we estimate that such a unit might be about 1 m (36-40 in.) in diameter and could be accommodated. With appropriate design, the height could be limited to fit into the roadway ponton.

We also estimate that the overall weight of one such unit would be about 1000 lb. In addition, there might be 250 lb of fluid in the unit and perhaps 200 lb of foundation structure plus 300 lb of fuel and fuel system, making a total of 1750 lb each or 3500 lb total (1591 kg). This, in turn, would increase the draft of the bay by 1.6 in. (4 cm) and would reduce the maximum lift capacity of the bay by about 4%, in a static situation.

The concept of an outdrive has not been examined extensively, because it would seriously violate the folded envelope of the bay. Provided an engine small enough to fit into the bow ponton could be found, and provided the structure of the bow ponton could be reinforced for the load, an inboard/outboard drive such as used on the Mobile Assault Bridge units could be installed under the rake of the bows. The engine and horizontal axis swivel would be to one side of the ponton. When "retracted," the unit would house horizontally against the undersurface of the bow ponton or the rake. Since it might be submerged even then, it would have to be suitably protected. By rotating it 90°, the unit would be lowered to about the level of the bottom of the bay. Propeller protection and increased thrust would be provided by a shroud or nozzle, but the unit would still be vulnerable to impact from debris. In deployed mode, access could be achieved from the walkway; in folded mode, the same access difficulties would be encountered with the other alternatives. Weight problems would be similar, and the prime mover would also be tilted 90° between the two modes. How much this extension outside of the folded envelope would affect transportability has not been examined closely. The unit would remain within the base width and height of the folded bay's cross-section but would protrude from the bow sections, tapering toward the top, and would raise the center of gravity. A sketch of a possible arrangement with gas turbine is provided in Figure 3-1.

Most notably, a Schottel unit in the roadway ponton or an outdrive in a bow ponton would raise the weight of the bay from 6.0 tons to somewhat over 7 tons (not including fuel and other fluids for the propulsion unit). The present 5-ton truck operates at reduced performance under the current overload. The question would arise whether the 5-ton chassis could be strengthened in the areas of suspension and drive train and axles to handle a 7-ton load at a reasonable cost--especially in view of the expected life span of these standard vehicles, which have been in inventory for close to 20 years--or whether such an increase implies a direct switch to the 10-ton 8 x 8 transporter, which will cost approximately $125,000 in 1982.

In summary, accommodation of the preferred Schottel unit in the roadway ponton or the outdrive unit in the bow ponton would require appreciable reconstruction of the ponton structure; installation would not be a simple backfitting job. The access question is most serious and does not appear to have a satisfactory solution which retains the structural and watertight integrity of the unit nor its readiness condition on short notice. The added weight problem carries over into both bay performance and transporter capability and selection. Introduction of self-propulsion into bays of current configuration and envelope requires considerable redesign of the



Source: ADL Dwg. SK-61881-2

Figure 3-1

Outdrive Concept for Bow Ponton of Interior Bay (Ribbon Bridge)

particular bay and enhances one capability at the expense of several others.

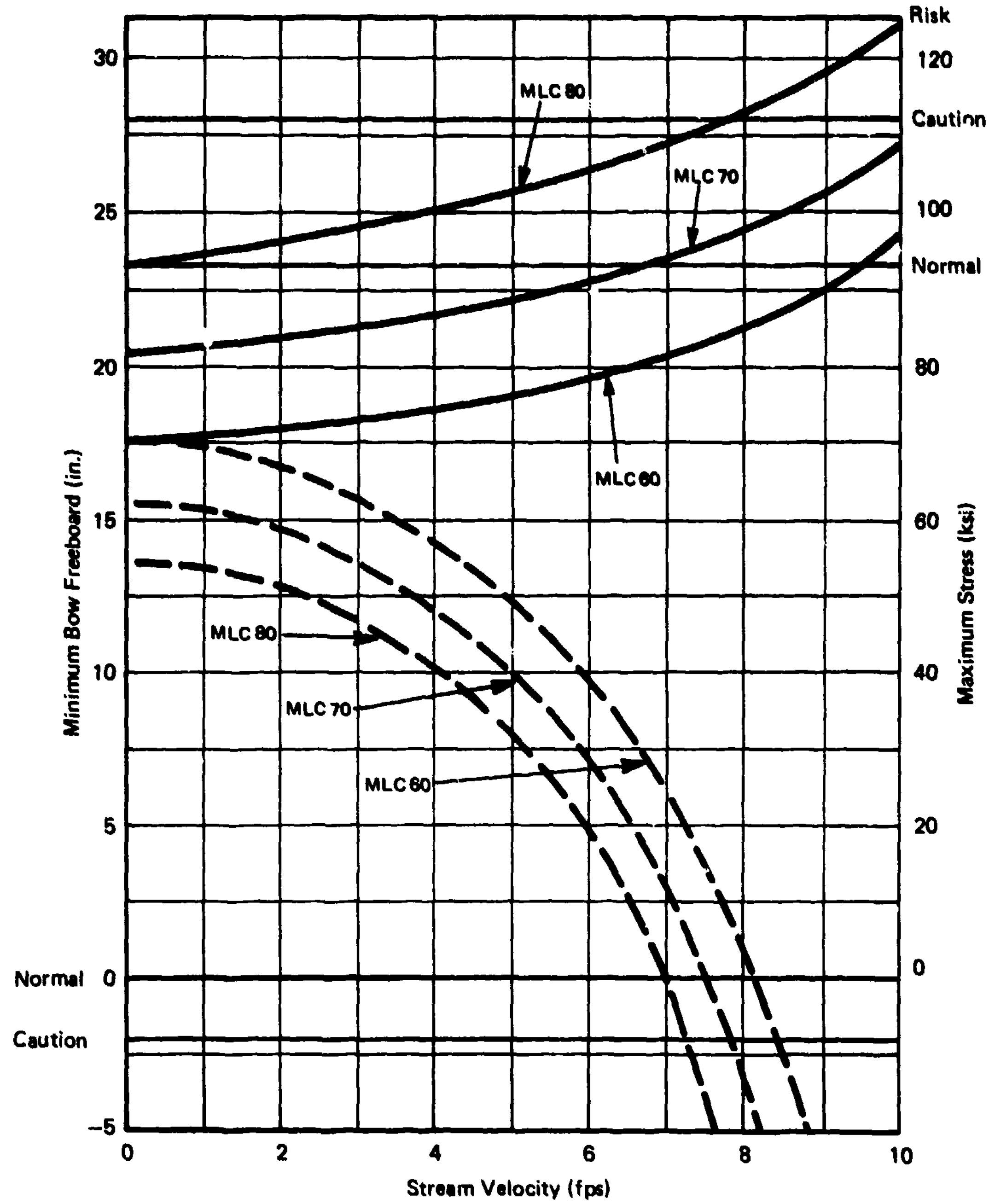
3.5 HYDRODYNAMIC PERFORMANCE

The capability of the present bridge is best illustrated by Figures 3-2, and 3-3, which represent data from recent field tests. With the introduction of the Abrams Tank, the bridge must be able to carry Class 70 loads. The remaining trade-offs concern any additional weights that might be placed in the bridge, which would further degrade its hydrodynamic performance.

The shape of the rake of the bow pontons is one of the factors that affect build-up of a bow wave in front of the bay. As the relative speed between water and bridge increases, this wave suddenly spills over the bow and suddenly thereafter the bow tends to dive. This phenomenon has long been recognized with towed or (particularly) pushed barges and has been found to be sensitive to angle of rake and shape of the bow. Note also in the bridge configuration the bays form a continuous structure of uniform cross-section; the flow past the bridge becomes almost two-dimensional. However, in the rafting mode the flow is more analogous to that about a ship and has large components in the third dimension, transverse to the principal direction of motion.

The shape of the bow pontons of the present Ribbon Bridge is tied to the folding scheme and to the transportation envelope. We do not see any simple way to alter the characteristics of the bow without interfering with those other aspects of the system.

For example, if the bow ponton were extended, even at the present angle of rake, it most probably would ride better, but its weight and

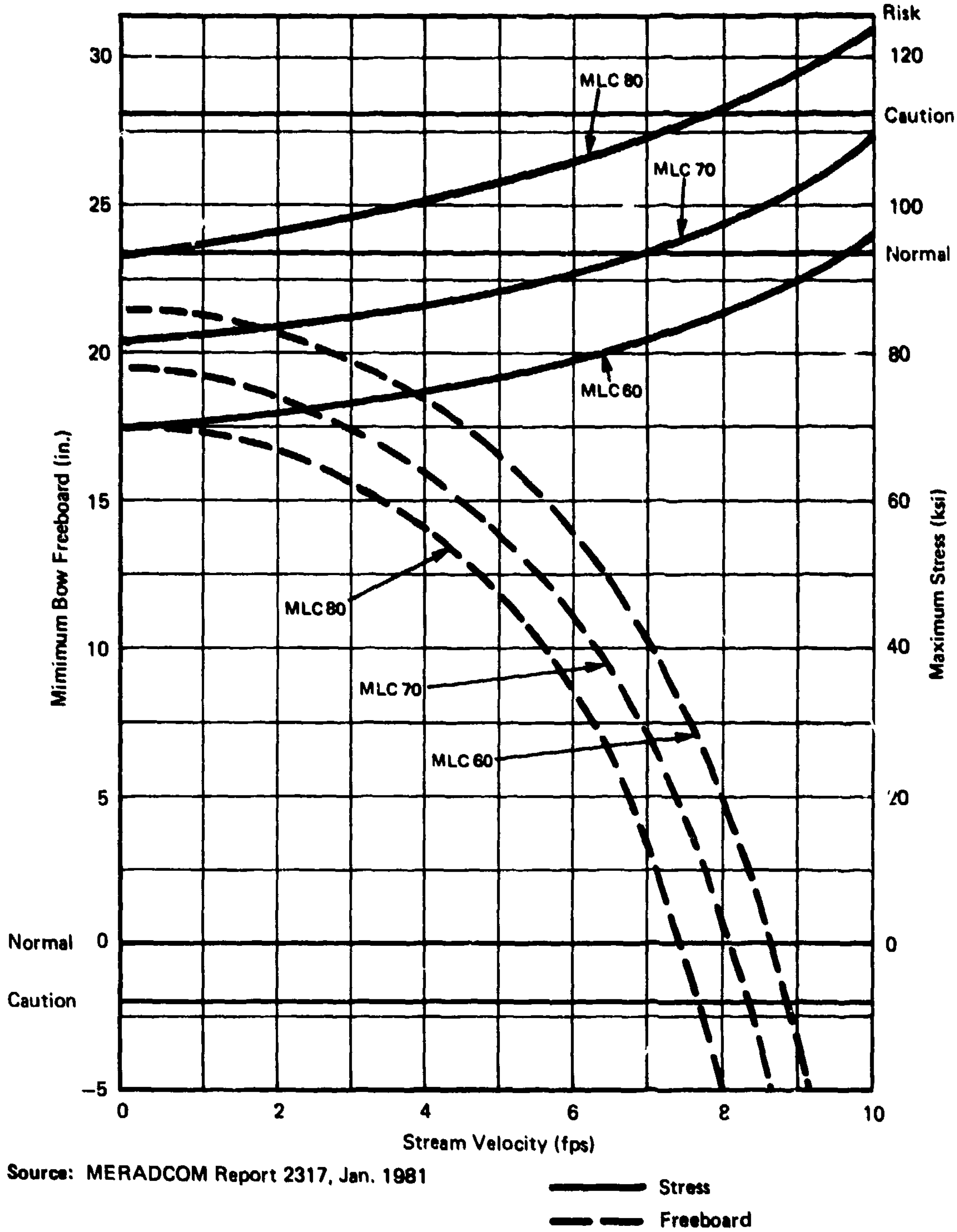


Source: MERADCOM Report 2317, Jan. 1981

Stress
Freeboard

Figure 3-2

Bridge and Raft Rating Curves for Dynamic Loading by Tracked Vehicles -- Load Fully Eccentric



Source: MERADCOM Report 2317, Jan. 1981

Figure 3-3

Bridge and Raft Rating Curves for Dynamic Loading by Tracked Vehicles -- Centered Load folded height would increase. If the angle of rake were decreased and the bow were extended in length, the situation most certainly would be improved, but the configuration limits would be changed entirely. If temporary extensions were used, such as collapsible bow bulkheads or inflatable bladders, some gains might be achieved, but performance degradation would be more abrupt after the velocity limits had been reached. Again, weight would be added as well as complexity and cost.

It is very difficult to predict the exact performance of different bow configurations, especially in the bridging mode, and full-scale tests are prohibitively expensive. Model tests, however, are reasonably simple, cheap, and rapid. The desired information could be obtained without the most sophisticated of towing tanks or the largest of models, and quite a number of institutions could perform such work.

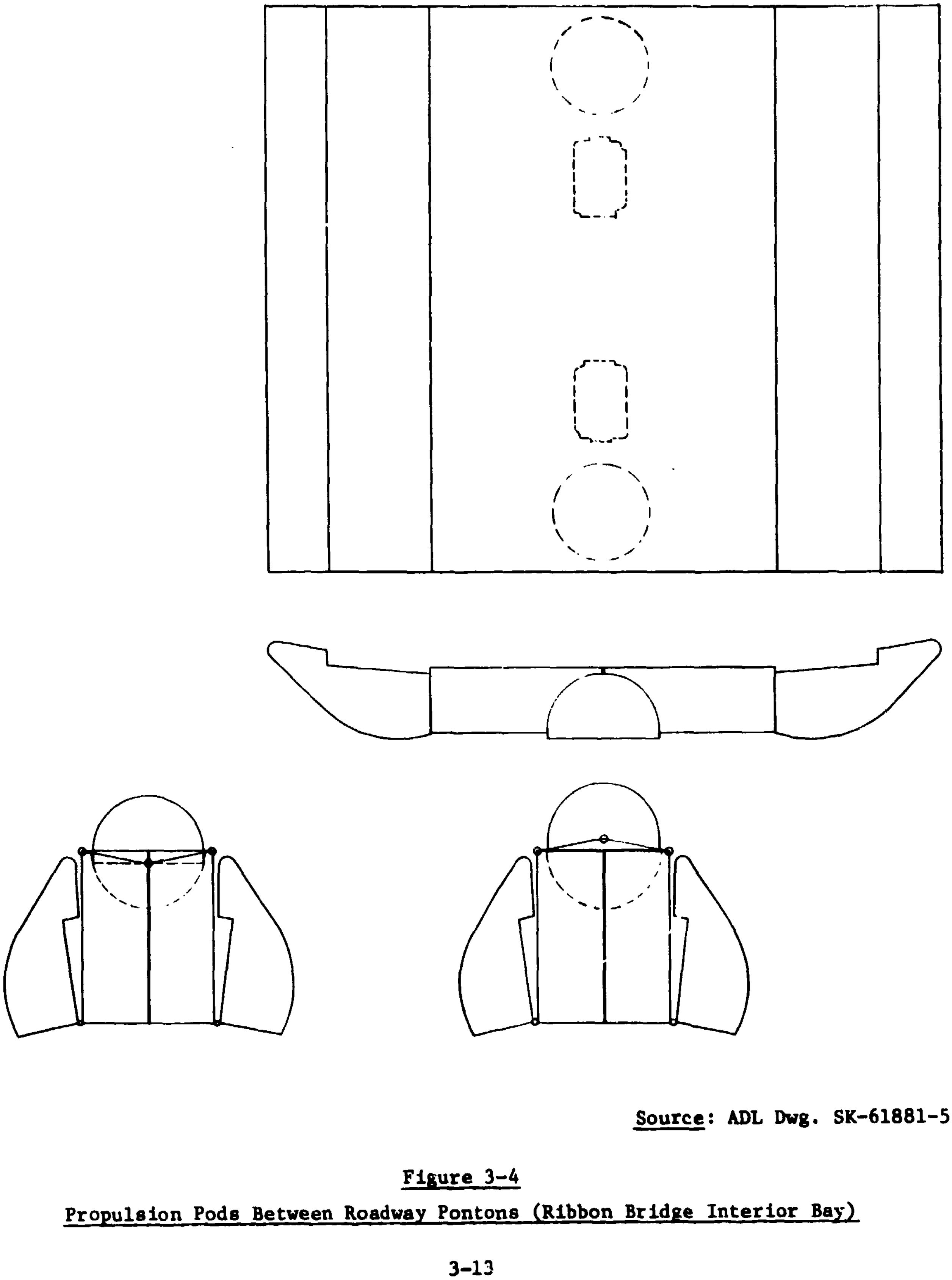
At this time, improvement of Ribbon Bridge capabilities appears to require a redesign effort. Our examination indicates that this would result in a unit somewhat different from the present bridge. The maximum overall dimensional requirement probably would change, which would affect the transportation envelope. The weight might also be greater bringing the transporter question back into the problem.

Inasmuch as the bridge will be required to operate under maximum load, the buoyancy of the individual pontons should be assured. At present, they are vulnerable to damage from debris or from grounding on hard objects on the river bottom, as well as to small arms fire. The bilge pump has limited capabilities and requires the attention of personnel. An obvious solution is to foam the inside of the ponton voids. The foam can be made fire-retardant. However, the problem of weight immediately arises again: foam weighs about two pounds per cubic foot, which would add about 1000 lb to the weight of a roadway bay. In a self-propelled unit with the engine inside the ponton, this weight would be proportionately less. Here also, the interrelation of bridge capability, reliability and safety, transportability, and survivability must be considered.

3.6 GIMBALLED PROPULSION PODS

We have previously mentioned the problem of having a diesel prime mover operating in one position and having to be stowed 90° from that position when the ponton is folded. This, together with the need for access in both positions, led to the consideration of using a gimballed propulsion pod, which would permit the engine and pump-jet to remain in their normal positions at all times.

In one version, (Figure 3-4) this pod is located between the two roadway pontons and contains both engines and pump-jets. Since its depth would be the same as that of the roadway pontons, an engine would have to be found that fit into this space. The geometry of the problem indicates that if the trunnions of this pod were in line with the hinge at the bottom of the two roadway pontons, the pod would project above the folded envelope a distance equal to the depth of the ponton. To minimize this projection, the trunnions or other supporting device would have to be movable so that the pod could be lowered into a void within the roadway pontons. While the pod itself would be a watertight enclosure, the void into which it would be lowered would be an open space, reducing the buoyancy of the bridge. The mechanism for lowering (or lifting) would add weight. The critical characteristics of bridge capacity and velocity capability would be impaired. Furthermore, this approach would require appreciable redesign of the roadway pontons, as contrasted to reconstruction or modification.



Source: ADL Dwg. SK-61881-5

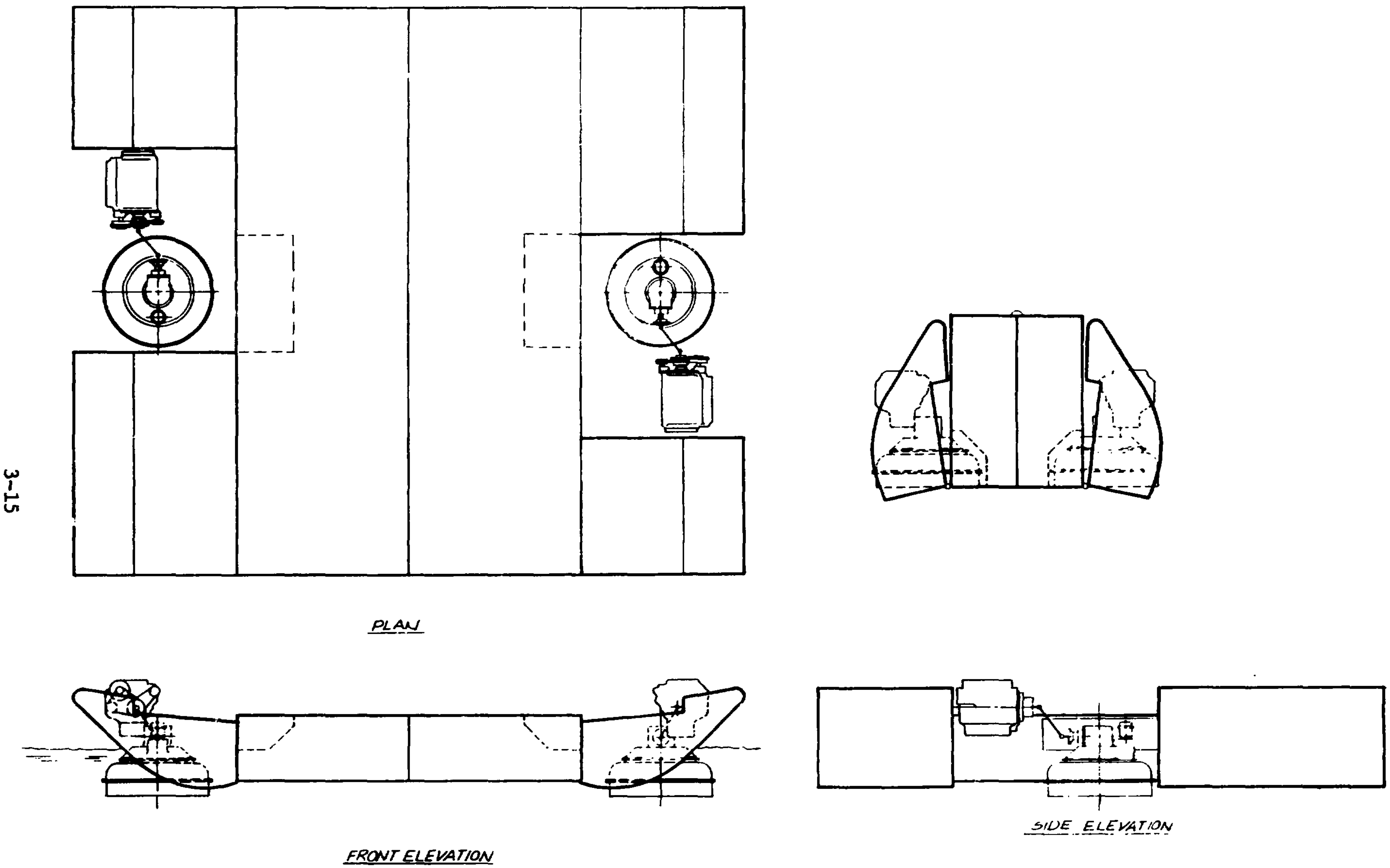
Figure 3-4

Propulsion Pods Between Roadway Pontons (Ribbon Bridge Interior Bay)

In a second version, two gimballed or rotating propulsion units are located in the bow pontons. Each bow ponton of the bay is recessed, and this recess would be closed off by appropriate bulkheads on either side. One propulsion unit in an appropriate enclosure would be fitted into each recess. Preliminary layout studies, as shown in Figure 3-5, indicate that the pump-jet would protrude from the rake of the bow ponton and could be fitted properly; that most probably a conventional diesel engine would require that a small area of the bow ponton walkway surface be raised; and that the diesel in rotated position would protrude from the rake of the bow when the bay is folded. Provided the latter is within the maximum width of the envelope and not beyond the maximum envelope height, this may be permissible. The weight problem and its relation to degradation of bridge performance as well as overload of the 5-ton truck transporter remains. Another factor, which can only be speculated upon at this time, is the folding and unfolding process with this added weight in the bow pontons. The dynamics of the unfolding place high inertial loads upon the components; since greater weight would increase these loads, the ponton structure as well as the propulsion system would require examination from this viewpoint.

3.7 LOAD/FREEBOARD/SPEED RELATIONSHIPS

A series of tests conducted during 1979 and 1980 are covered in the report "Ribbon Bridge Rating Test for Bridge and Raft Configurations", Report 2317, by Dan Causey, Jr., and Carlos A. Piad, MERADCOM, January 1981. In summary, the report concludes that the bridge is capable of carrying a Class 70 load in a current of 7 ft/s; a 5-bay conventional raft can operate in a current of up to 8 ft/s with an MLC 70. The bridge can



Source: ADL Dwg. SK-61881-6

Figure 3-5

Gimballed Pump-Jet in Bow Ponton (Ribbon Bridge Interior Bay) operate with some sections submerged under a traveling load, since its torsional strength provides stability; rafts, on the other hand, are more sensitive to bow waves on deck, and a risk situation is quickly reached under this condition.

The following sentences are quoted from page 69 of the report: "The result of the bridge testing, raft testing, and correlation of all data on the EDT (Engineer Design Test) results was a determination that the Ribbon Bridge has sufficient freeboard to sustain the MLC 70 loads with no negative freeboards up to 7 ft/s. The bridge would begin to lose capacity at this current, and the freeboard would control the rating from this point."

Graphs from Appendix R of the above report (shown here as Figures 3-2 and 3-3) illustrate the load/freeboard/speed relationship. Note the steep slope of the curves beyond 4-5 ft/s and the increasing decrement of freeboard.

To relate the propulsion problem to these curves, if each bay has a lightweight increase of two tons due to the installation--or, conversely, if it loses two tons of load-carrying capability out of about 20 such tons--the total loss for a five-bay raft would be 10 tons. This is the equivalent of a change between an MLC 70 and MLC 80 load. From Figure 3-2, proceeding from the rated 6 ft/s for the MLC 70 to the MLC 80 curve at constant freeboard, there would be a 0.6 ft/s reduction in stream velocity. Thus, at this point the 10% reduction in load-carrying capacity of the raft would be accompanied by a 10% reduction in the permissible water speed.

3.8 STATIC CAPACITY OF INTERIOR BAYS

Although Figures 3-2 and 3-3 are plotted in terms of minimum bow freeboard versus stream velocity, some salient considerations become apparent from the static characteristics of the units.

The light draft, from the lowest point of the bow ponton of an interior bay, has been calculated by us from the shape and dimensions to be 8 inches at a lightweight of six tons. This agrees with the provided information and confirms the values used in our computation.

If the maximum permissible draft to the bottom of the main roadway structure is 30 inches, the bay has a deadweight capacity of 28.1 ST. If the maximum permissible draft is 32 inches, which would be within one inch of the upper roadway surface, the capacity rises to about 31.8 ST. Three bays would be required to support a balanced MLC 70 statically with dry roadway surface.

An increase in weight of about 1.75 ST for a two-propulsor installation reduces the deadweight capacity of the bay to 26.3 ST and 29.3 ST for the two drafts respectively, which might make a fourth bay necessary if a full 70 tons had to be supported. Since neither equal distribution among the bays nor exact positioning and balancing are to be expected, a lower capacity must be used to allow a safety margin.

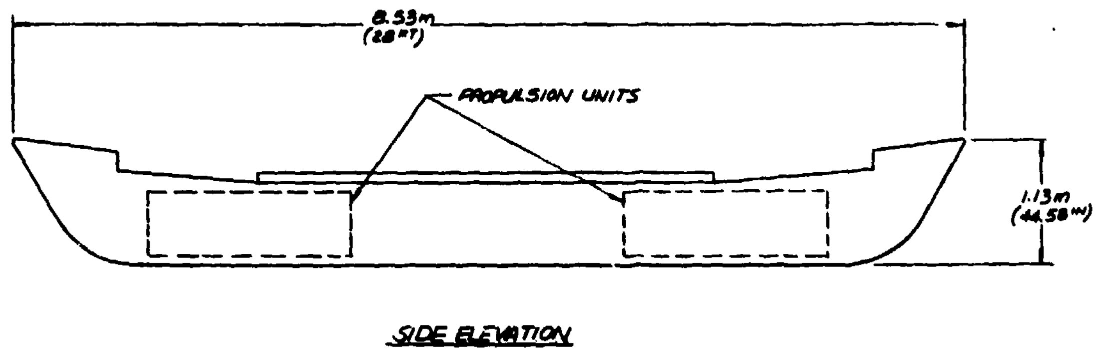
Foaming of the bays with material at two pounds per cubic foot would add approximately 0.5 ST to the lightweight per day; this would reduce the deadweight capacity commensurately or increase the draft by 0.4 in. at the load deep draft.

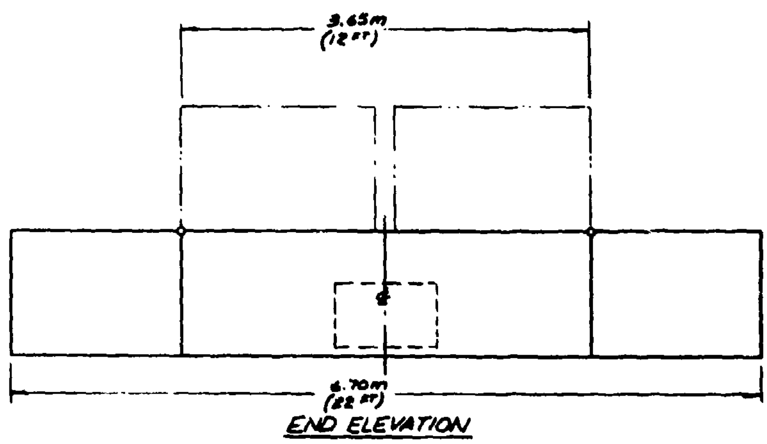
3.9 PREFERRED ALTERNATIVE (FIRST CONFIGURATION)

The several improvements examined so far involve either extensive redesign or a weight increase that penalizes as much as it benefits the system. As an expedient which does not interfere with the characteristics of the Ribbon Bridge as now configured and which might be considered as an addition that partially solves the problems, we suggest a separate powered unit. This differs from a Bridge Erection Boat in that it can be integrated into the bridge or ferry as well as used independently. The difficulties that emerged during our search for ways to fit integral propulsion into the pontons of the standard Ribbon Bridge suggested the concept of a separate self-propelled unit which would fit into the Ribbon Bridge system but which might avoid some of the problems of space, buoyancy, transport weight, and engine access and psoition. Figures 3-6 and 3-7 are sketches of some early ideas; both were discarded because they provided insufficient benefit to justify exceeding the transport length limit.

A more practicable concept is shown in Figure 3-8. This unit is fully powered and steerable. It is transportable on the standard M812 5-ton chassis and falls within the transportable envelope of the bridge units. Its draft is comparable to that of the bridge units, and it provides most of the functions of a Construction Boat.

To provide sufficient buoyancy, and also to fit into the road envelope, the hull must be sectionalized. In this design, however, only the bow ponton is hinged; the roadway ponton is in one piece. When unfolded, the distance between the tips of the bow pontons will be greater than in an interior bay, but the unit will be only about half as large in the other dimension. For propulsion, two Schottel Pump-Jets are proposed. In profile, the unit will resemble an interior bay with an extended bow and stern. The hinge and locking mechanisms will match those used on standard interior bays; the protruding pads and pins may present some design problems and inefficiencies but are a reasonable trade-off for the utility of the craft.

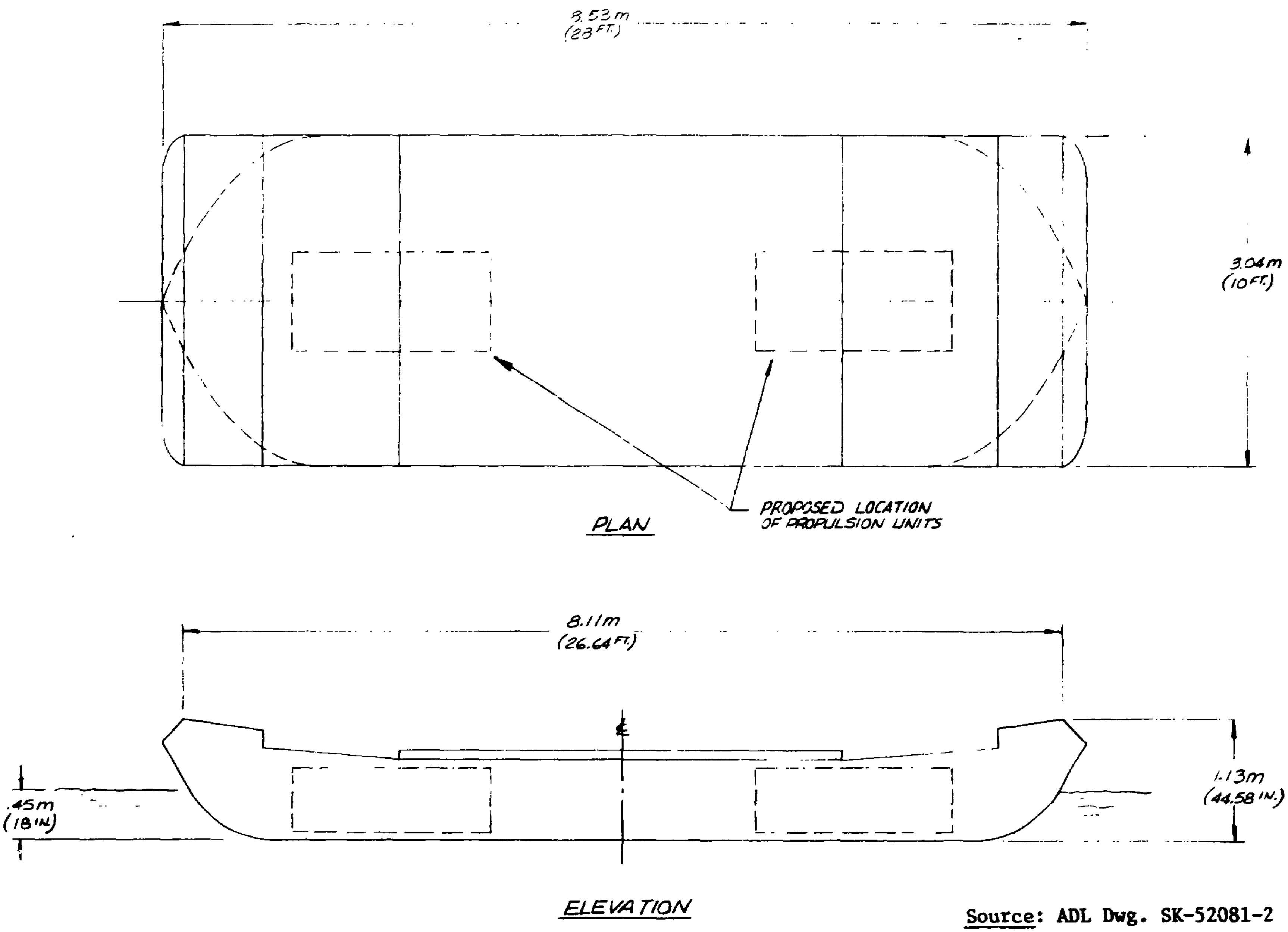




Source: ADL Dwg. SK-52081-3

Figure 3-6

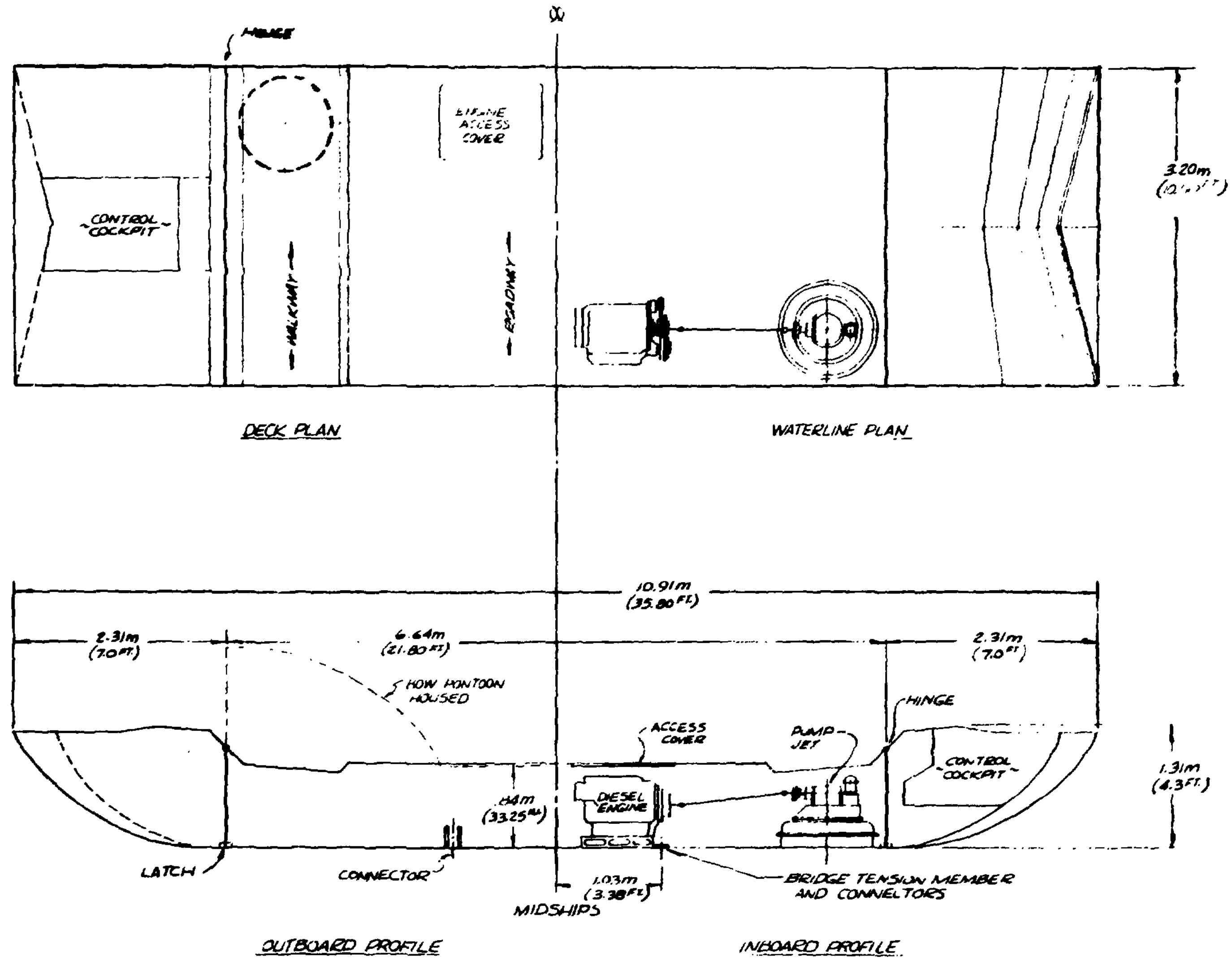
Three-Part Integral Propulsion Interior Bay for Ribbon Bridge



Source: ADL Dwg. SK-52081-2

Figure 3-7

Single-Ponton Propulsion Bay for Ribbon Bridge/Raft



Source: ADL Dwg. SK-61881-1

Figure 3-8

Propulsion Bay with Folding Bow Pontons (Half-Bay) for Ribbon Bridge/Raft

The design of the protruding bow and the method of folding should be noted. The single-ponton mid-body or roadway ponton is approximately 22 feet by 10 feet and houses the two propulsion units. The two bow pontons, which are buoyancy units except for the space occupied by the operating/control stations, stow on top of the roadway ponton. Since each of these hinged pontons is 7 ft long, the extended length of the bay or boat is 36 ft; in the folded mode a clear deck space of 8 ft on the roadway surface permits access to the propulsion units.

A bow protruding from a line of standard bays will tend to deflect a bow wave to either side, thus increasing the bow wave on the standard bays and aggravating this critical situation. Even a bow with a low rake would have this tendency, and a bow with a finite entrance angle to the waterplanes obviously generates a flow with an outward component. We therefore suggest further investigation of what is known, among various names, as a tunnel bow, which consists of an inverted V-shaped bottom at the after end of the bow ponton, where it mates with the flat-bottomed roadway ponton. Moreover, such a bow can be designed with developable surfaces--that is, with curved surfaces that bend in only one direction. These can be formed on plate rollers and do not require the heating and forming used for double-curved surfaces. Such shapes are within common steel fabrication capabilities. Figure 3-9 indicates how these surfaces are designed and the types of curves that result. A flat-surface, constant "V" also could be used but would result in a sharp knuckle at the baseline. A combination of the two shapes would also be possible.

This type of bow has two additional advantages: (1) the two fore feet provide stability if the craft is grounded head-on against a bank, and (2)

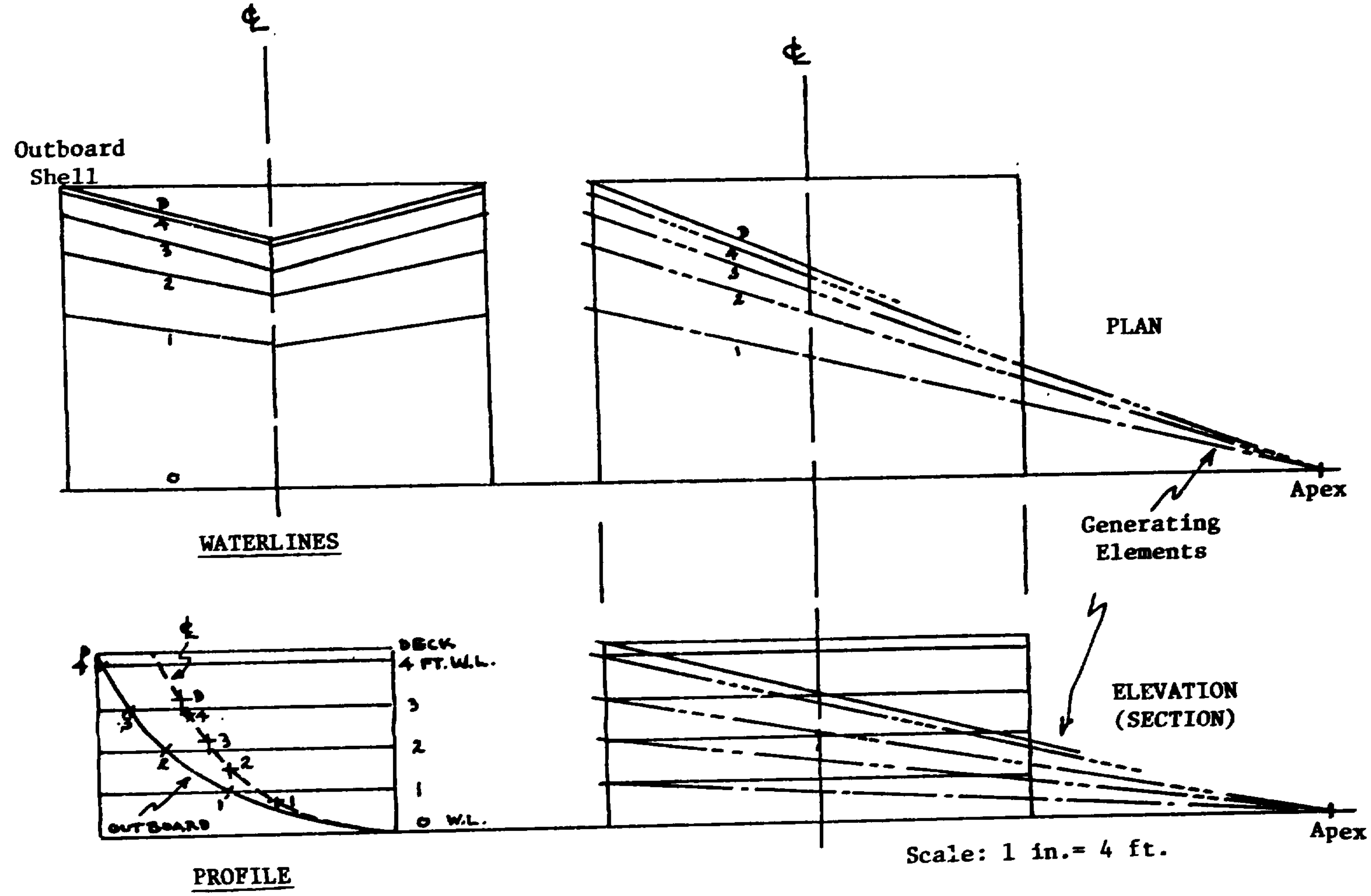


Figure 3-9

Generated Surface of Tunnel Bow

the width at deck level provides a good bow shape when the craft is used as a pusher tug for other floating equipment. The best lines for the inverted "V" would have to be developed by model tests.

The simplest method of articulating the bows would be to hinge them at deck level, as in Figure 3-10. From first estimates of the dimensions and geometry of such an arrangement, unfolding of a bow would require that its center of gravity be raised about two feet, after which it would drop about four feet. Some of this energy input and output can be equalized by use of a torsion bar or torsional spring.

Another approach would be to keep the bow section horizontal and let it be supported on trunnions and arms, with secondary free arms to form a parallelogram as in Figure 3-11. The energy relations remain the same. While the bow section or the trunnion arms rotate 180° in either scheme, the peak elevation of the center of gravity of the bow section does not come at the 90° point; therefore it is not feasible to balance the system completely by a torsion bar.

If the bows are extended after the unit has been launched, an additional energy requirement appears. The midbody initially will float at a draft greater than the extended craft. As the bows are unfolded, energy must be expended to depress the bow sections and to raise the midbody until all three sections float at a uniform draft. During retraction of the bows this energy could be recovered, since this is a reversible process. Nevertheless, it complicates the extension process in water. The quantities of energy in and out during each part of the cycle can be calculated precisely once the dimensions and weights of the sections of the craft are determined. Further explanation of this calculation is given in section 3-10.

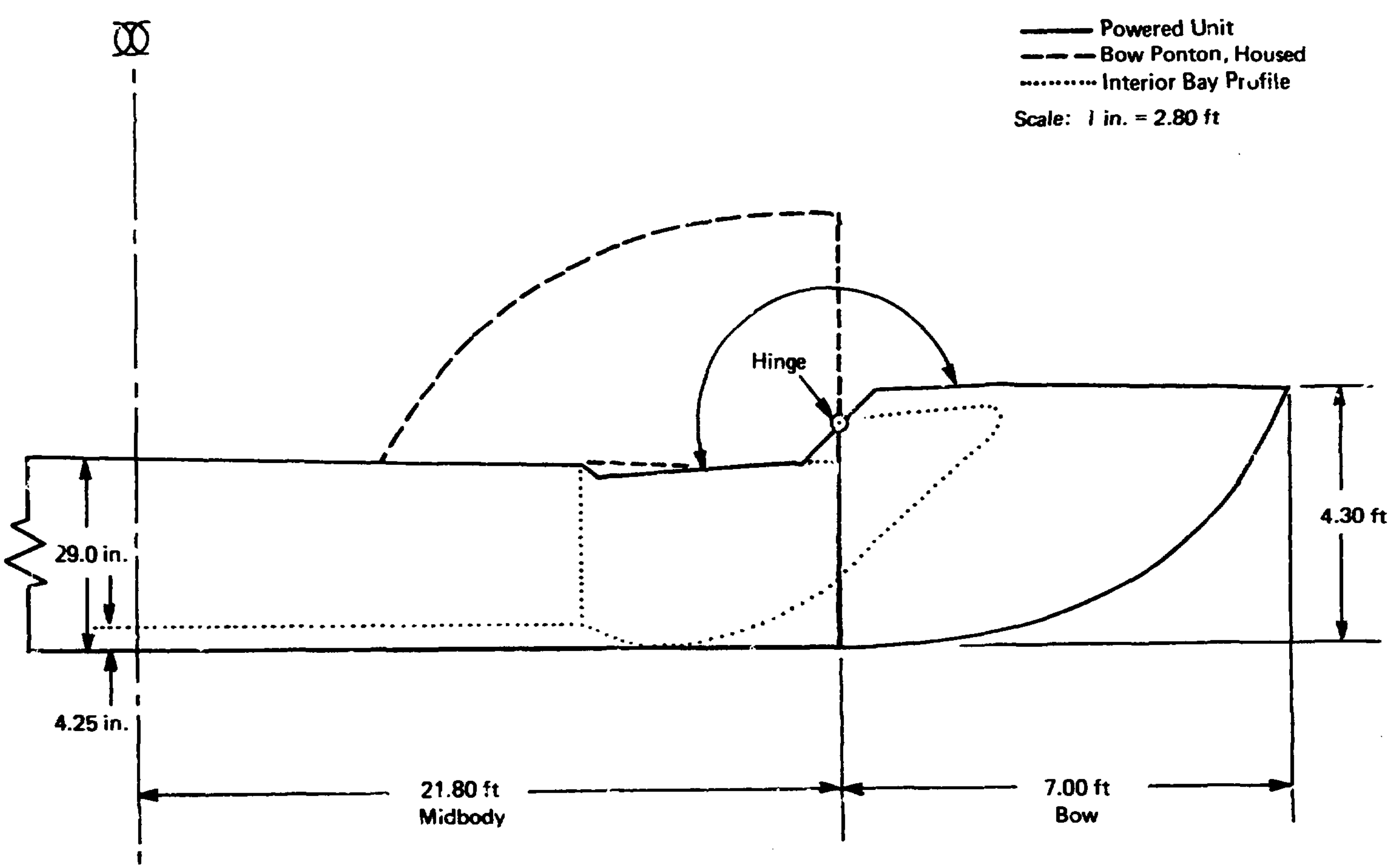
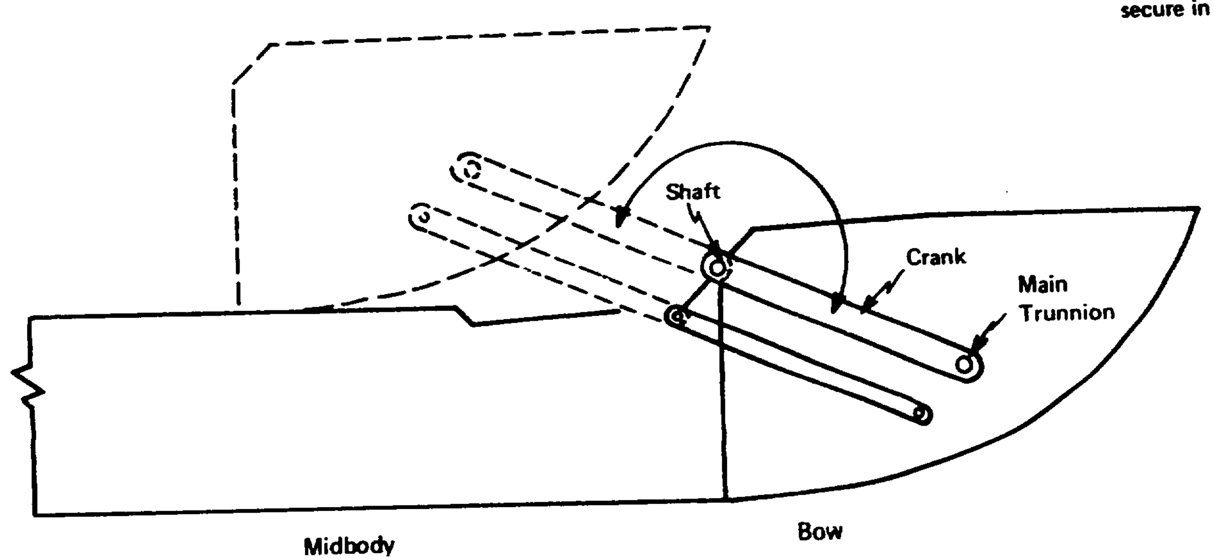


Figure 3-10

Hinged Bow Ponton in Stowed and Deployed Modes



Note:
Although shown the same as in the hinged arrangement, the bow profile has more latitude of shape in this stowage but may be more difficult to secure in the retracted mode.

Figure 3-11

Bow Ponton Supported on Trunnions and Arms

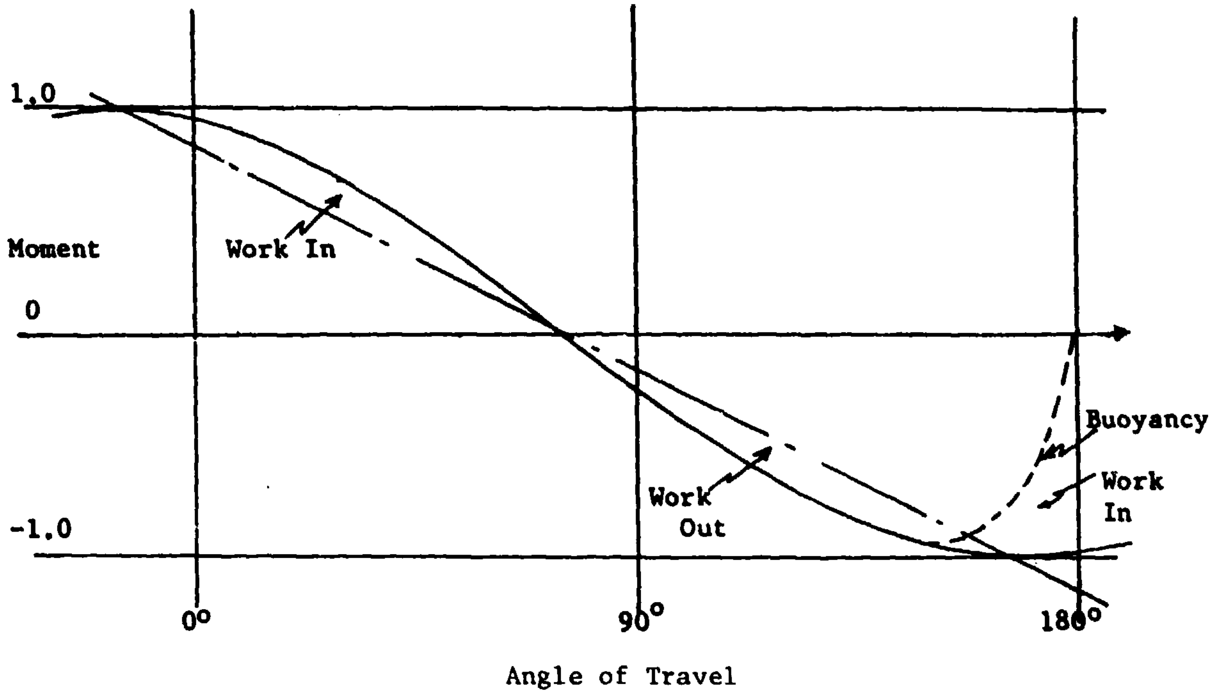
In the hinged case, a shaft drive geared to a hydraulic motor might be appropriate, with a hydraulic pump fitted to one of the main air-cooled diesels. The hydraulic system, if fitted with an accumulator of sufficient size, also could serve as an energy storage device. Cable operation of the folding/unfolding mechanism is feasible, but the geometry indicates that protrusion above the deck would occur, with attendant increases in vulnerability and obstruction to traffic. A hydraulic-mechanical or electromechanical drive still would be required.

In the trunnion arrangement of Figure 3-11, a geared hydraulic motor drive likewise is indicated. In both cases, hydraulic hand pumps can be installed for emergency use when the main engines are inoperable. Direct hydraulic piston actuation may be feasible, but the 180° movement and the nature of the craft suggest that it would be extremely difficult to fit the pistons internally in a protected location.

From preliminary layouts, the hinged approach appears to provide a simpler and more secure seating arrangement in the folded mode. The principal concern in this scheme is restraint or control of the bow section as it drops from its highest elevation. Impact with the water or the midbody could generate destructive inertial forces. For deployment in water, power must again be supplied near the end of the travel to complete the motion of the bow section.

3.10 ENERGY RELATIONSHIPS IN BOW DEPLOYMENT AND RETRACTION

The moment exerted by the ponton at the hinge due to gravity depends on its weight and the position of the center of gravity (its distance from the hinge and the angle a line joining the two makes to a horizontal reference). At 0° or 180° this moment is maximum, while at the 90° point it is 0; this is a simple cosine relationship (Figure 3-12).



Legend

——————— Ponton, by gravity

- ———— - —— Provided by torque bar

- - - - - - - - Buoyancy in water

Maximum moment, M = Wd = 1.0

where

W = Weight of bow section

d = Distance from hinge to center of gravity of bow section

Figure 3-12

Typical Energy Relationships in Bow Deployment and Retraction

In the stowed position the initial angle between the horizontal and the hinge-CG line is at some value greater than zero, and in the fully extended position the angle is greater than 180°. There is a constant phase-angle difference between angle of travel of the bow section and the cosine function of the moment.

From the stowed position at 0° travel to the zero-moment point, the area under the sinusoidal curve represents work which must be put into the system to raise the ponton to its upper netural position. The analogous area to the right of the zero-moment point represents work which the ponton releases as it descends to the extended position under the force of gravity.

A torque bar or linear torsional spring at the hinge can be set to have the effects shown, i.e., to exert a torque equal to the maximum moments at the same angular points and zero torque at the zero-moment point. The energy input and output quantities are greatly reduced as the spring stores the energy required and released by the ponton.

If the bow is extended in water, its buoyancy will necessitate additional energy input at the end of the extension travel (indicated by the dashed line in Figure 3-12). During retraction or stowing, the energy inputs and outputs are reversed.

To illustrate the approximate values, assume the bow ponton weighs 1500 pounds and is 7 feet long and 2.5 feet deep, with the center of gravity about 3.72 feet from its hinge and 19.65° above the horizontal from the hinge. The maximum moment (at an angle of travel of 160.35°) would be 1500 x 3.72 = 5580 ft-lb. The energy requirements are derived by direct integration of the curve: the area from 0° to 70.35° travel requires an input of 3701 ft-lb, while from that netural point to the seating at 180° returns 7445 ft-lb.

With the counteracting effect of a torsion spring (diagonal line in Figure 3-12), the initial energy input between 0° and 70.35° is reduced to 1026 ft-lb and there is an energy return of 1198 ft-lb between 70.35° and 160.35° of travel.

From 160.35° to seating at 180°, a small energy input of 245 ft-lb is required when the system is deployed in air. However, when deployed in water, this final increment increases to approximately 1200 ft-lb; the exact value depends on various characteristics of the center and bow pontons.

For the 1500-pound ponton with a 7-ft by 2.5-ft profile and a torsion bar spring, the following matrix sums up these results:

	Deployment (Read Down)		
	Energy In	Energy Out	
In Air:	1271	1198	(foot pounds)
Afloat:	2226	1198	
	Energy Out	Energy In	
	Retraction (Read Up)		

3.11 INTERCONNECTION

One of the complications which arise from this concept is that although the standard interior bays and the powered units have different light weights and different waterplane areas (or tons per inch immersion), their drafts must be equalized when the two are to be connected.

The standard interior bay draws 8.8 inches in light condition. The powered unit with dimensions as indicated and a weight of 5.5 ST draws 7.8 inches. (Even in folded mode, it draws only 9.3 inches.) At 15.8 inches draft the unit displaces 11.5 ST. If the unit were to be connected to one or more interior bays, its draft would have to be increased to 8.8 inches by addition of 0.66 ST of ballast. If this weight could be removed after the units are connected, it would provide additional lifting power for the bridge or raft.

Ballasting could, of course, be avoided by altering the proportions of beam and draft of the powered unit; however, its dimensions are essential to the system and should not be changed. A simpler solution lies in equipping the powered unit with ballast tanks large enough to increase the draft to the required 8.8 inches. In fresh water, one short ton equals 32.14 ft^3; therefore, tanks with a volume of 22 ft^3 (165 gal) would be required. To permit rapid ballasting and deballasting, a small pump would be required. Even if the pump were driven off the main diesels, the piping, valving, and pump would add several hundred pounds to the weight of the unit.

When empty, the ballast tanks would serve as watertight voids, and the ballast pump could be used as a bilge pump in case of damage. The variable ballast feature permits matching the draft of the unit to that of the standard units or varying the extra support given by the powered units to the array.

3.12 PREFERRED ALTERNATIVE (SECOND CONFIGURATION)

The conceptual design of the powered unit was developed under the assumption that broadside movement of rafts was the conventional mode. Since the most severe flow velocities in this case coincide with the flow directions encountered by the roadway bays when emplaced in a fixed bridge, the bow sections of the proposed powered units were, on the one hand, removed from strict lengthwise constraints, but on the other hand, required special treatment to prevent further interference with the non-powered bays. These projecting bows were desirable to increase bouyancy and to provide additional volume and deck space for operational and control functions.

However, it now appears that longitudinal rafting would be more desirable; the earlier emphasis on broadside rafting originates from behavior of the equipment rather than from operational reasons. Accordingly, the first configuration of the powered units is not sufficiently flexible. To permit either mode of rafting (and, of course, bridging),we have examined a second configuration of the concept that entails penalties of less buoyancy and less deck space. The salient features of this configuration are as follows:

a. Its profile is identical to that of the extended Ribbon Bridge interior bay, 8.16 m (28.6 ft) in overall length, with two exceptions:

   - The draft of the midbody is carried to the full depth of the bottom of the bulge on the Ribbon Bridge bay bow pontons--that is, the depth of the midsection is increased by about 4 inches.
   - One walkway is raised about 4 in. to accommodate a hinge line and a buried or hidden hinge mechanism.

b. The craft has one retractable bow only, of about 1.43 m (4.7 ft) length, which brings the stowed length of the unit within the permissible road transport length and is analogous to the 7-m length of the Ribbon Bridge interior bay.

c. When connected to one or more Ribbon Bridge bays, the craft will present continuous roadway and rake surfaces. The 4-inch bottom difference will cause some additional drag when in the longitudinal rafting mode, but this can be alleviated by rounded bilge chines.

d. The power source for a full-powered craft, i.e,, one with thrust equal to that of a Bridge Erection Boar or sufficient to propel two interior bays under worst conditions, remains the most severe problem. When the Schottel Pump-Jet (SPJ-50) is used, about 150 hp is required to produce this thrust at 2300 rpm. The further changes which result from this problem are discussed in later paragraphs.

With this smaller craft, it is desirable to consider the beam in the light of two factors:

a. Its draft must be slightly less than that of the interior bays when the craft is launched, so that it can be ballasted, if necessary, to match. If it has a greater draft, the connecting process would become impractical.

b. The proportion of length to beam must remain within a reasonable range in order for the craft to remain maneuverable when operating as a boat.

Assuming that the boat on launch (with fuel on board) has a total weight of 6.0 ST and a beam of 10 ft, it would draw about 11.7 inches. Some 4000 lb of extra buoyancy would be required to reduce the draft to 8.8 in to match the light launching draft of the interior bay. Extrapolation indicates that with a beam of 15 ft, the boat would have a draft of 8.8 inches. However, transport restrictions limit the maximum beam to 11.5 ft, which would require 2800 lb (1.4 ST) of additional bouyancy to provide an 8.8-in. draft. This could be achieved by lowering the bottom 3.5 in. further--that is, increasing the draft of the boat 7.5 inches beyond the flat bottom of the interior bay. When submerged to the bottom of the roadway, such a craft would have a deadweight capacity of about 12.9 ST on 11.5-ft beam or 12.0 ST on an 11.0-ft beam.

For geometrically similar shapes, structural weight may be estimated to vary as the square of a linear dimension or, in the case of floating craft, as the 2/3 power of displacement. In the case of this particular craft, where the profile is almost identical, two of three dimensions are similar and structural weight would vary as the beam of the craft.

Assuming an 11-ft beam, a preliminary weight estimate can be made as follows:

Proportion of width (in feet): 11.0/22.7 =	.4846
Proportioned weight (in ST): 6.0 x .4846 =	2.9075
10% increase for added longitudinal strength:	.2908
Powered hinge, less 2 passive hinges, 500 lb:	.2500
Hull subtotal	3.4483
2 sets propulsion (PJ + diesel @ 2000 lb ea.):	2.00
Fuel (50 gal @ 7 lb/gal ea.)	0.35
Control and navigation:	1.15
Total	6.9483 ST
Margin	0.0517 ST

The greatest interior height within the standard interior bay is 26 inches. The probable height within the proposed configuration is 30 in. and may be 33 inches. Air-cooled diesel engines in the 150-bhp range are at least 34 in. high. Even the liquid-cooled 90° V-8 Caterpillar engine, Model 3208, is 33.6 in. high. It is evident that if a diesel of any kind is to be used, either a portion of the deck has to be raised or a portion of the bottom has to be lowered. Even the 33-in. (or more) depth necessary to mate with the connecting interior bay would not fully suffice.

Gas turbine prime movers provide little net savings of space or weight, because they require some kind of variable-speed transmission. Their specific fuel consumption is also at least twice that of a diesel, their cost is high, and they raise problems of clean air intake and hot gas exhaust. Therefore, gas turbines are not considered further.

The water-cooled diesel with a closed cooling system and a heat exchanger connected to some form of hull-mounted cooler adds the weight of the hardware and the liquid. Under the tight weight limits, the latter is undesirable. Furthermore, complexity and vulnerability are added to the propulsion system. The advantage of such a system is that the engine can be operated briefly on land. With an open system that draws cooling water directly from the stream, the engine cannot be operated on land, but the transport weight is that of the dry engine alone.

In view of the probable requirement for increasing the depth of the craft due to the interbay connection problem, the water-cooled diesel loses further attraction. The high beam of the craft would not be reduced by use of a water-cooled diesel.

The air-cooled six-cylinder Duetz engine Model BF6L913 is a turbocharged engine producing 150 bhp at 2300 rpm; thus, it would not require a reduction gear when driving the SPJ 50 Schottel unit. Its length is 44-1/4 inches and its height is 36-1/4 inches. Within this height is included a deep oil pan or sump,which mightbe made shallower to reduce the overall height.

The height available inside the proposed hull is 30 in., possibly more. On top of this there is a 3-in. thickness of roadway extrusion. A 4- or 5-in. elevation above the upper surface of the roadway would make the available height at least 37 inches. Since the engines are 44-1/4 inches long mounted side by side, a 4-ft. long section across the width of the boat would provide the needed space while giving vertical vehicle (axle or hull) clearance, and it would be sufficiently narrow for small wheeled vehicles to pass or straddle the section. The raised portion therefore would be 4 feet (or slightly less) in the lengthwise direction of the boat and engines and would run across the full width of the craft. The vertical sides of this rectangle would act as a coaming to prevent water entry. The 3-in. thick roadway extrusion would not be needed; a thinner grill which also provided air entry and escape could be incorporated. The raised deck section would also be made portable and serve as an engine hatch. Figure 3-13 illustrates this configuration.

These estimates have been made to examine the feasibility of the concept and its major constraints. They indicate that the weight of the craft can be kept within the 6-ST limit while providing the desired power and maintaining a draft, in launch condition, similar to that of the bays with which it must operate and connect. Whenever this craft is operating with Ribbon Bridge units, it would be connected to them by the normal connecting pads and pins. Only in immediate launch and retrieval situations might pushing be required; for that case, knees could be fitted at the bows to facilitate such operation.

Figure 3-14 shows the integral propulsion half-bay for the Improved Ribbon Bridge on the current Ribbon Bridge transporter (the M14, 5-ton truck chassis) with the single folding bow ponton in its folded or stored position.

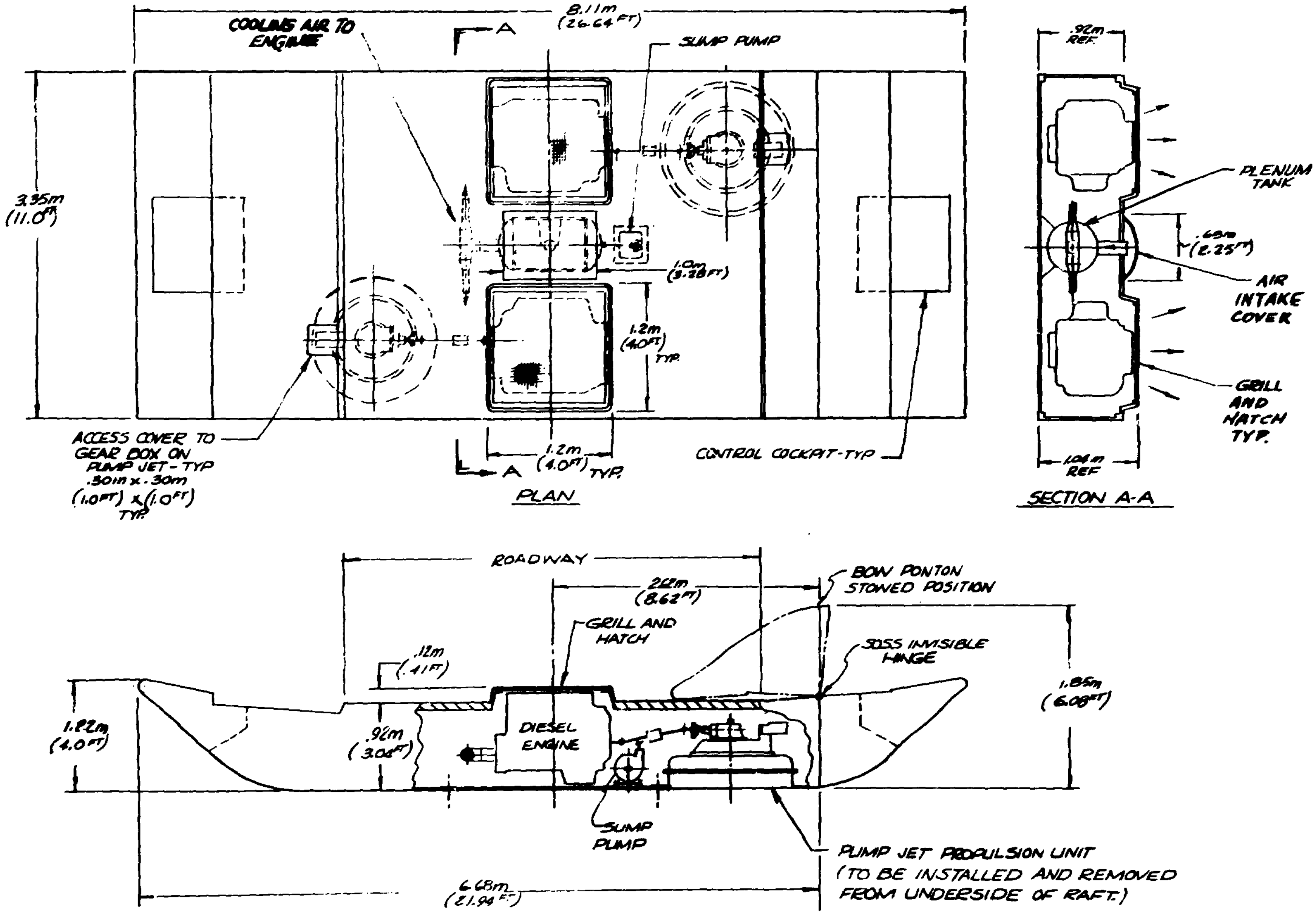
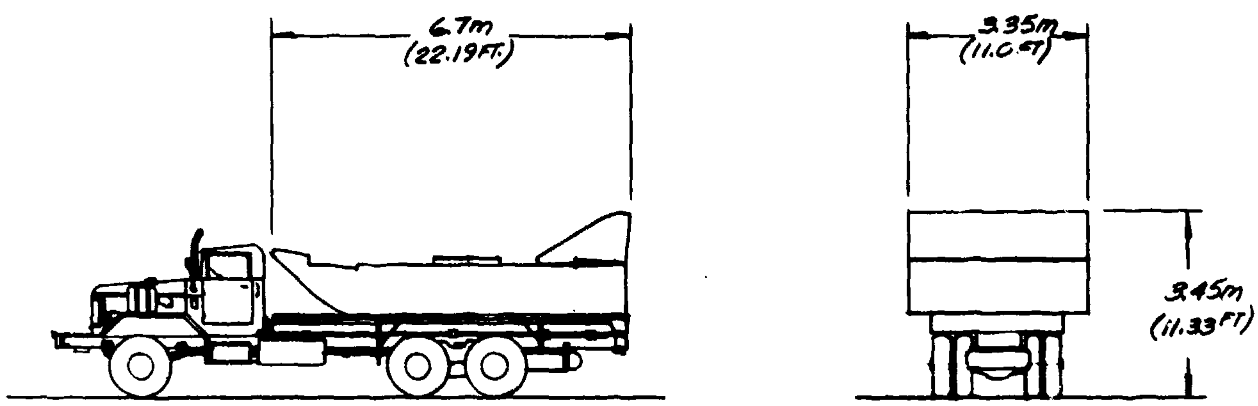


Figure 3-13

Source: ADL Dwg. SK-61881-10

Propulsion Bay with Single Folding Bow Ponton



(ADL Dwg. SK-61881-14)

Figure 3-14

M814(6X6) 5-Ton Cargo Truck (Proposed Transporter for Ribbon Bridge/Raft Propulsion Bay)

4. THREE-PONTON BAY SYSTEM WITH INTEGRAL PROPULSION

In the amended Statement of Work, the study of the Three-Ponton Bay System, as described in Figures 1, 2, and 3 of the Statement of Work, is to consider two alternative integral propulsion subsystems--one with a pump-jet and the other with an outboard drive. Both of these subsystems have been reduced to Level 1 drawings. The pump-jet drive utilizes a diesel prime mover; the outboard drive utilizes a gas turbine. The envelope of the three-part interior bay was specified in the Statement of Work.

The Statement of Work also required consideration of providing integral power in the ramp bays. Because of height restrictions, dual propulsion systems cannot be accommodated in these bays, but a single pump-jet could be installed. For control, the ramp module must use its own intertia as a reaction moment. Hence, as a launch procedure, we recommend that an interior three-part bay be launched first; the ramp bay should then follow and be joined to the interior bay, so that improved control can be obtained with the dual integral propulsion of the interior bay.

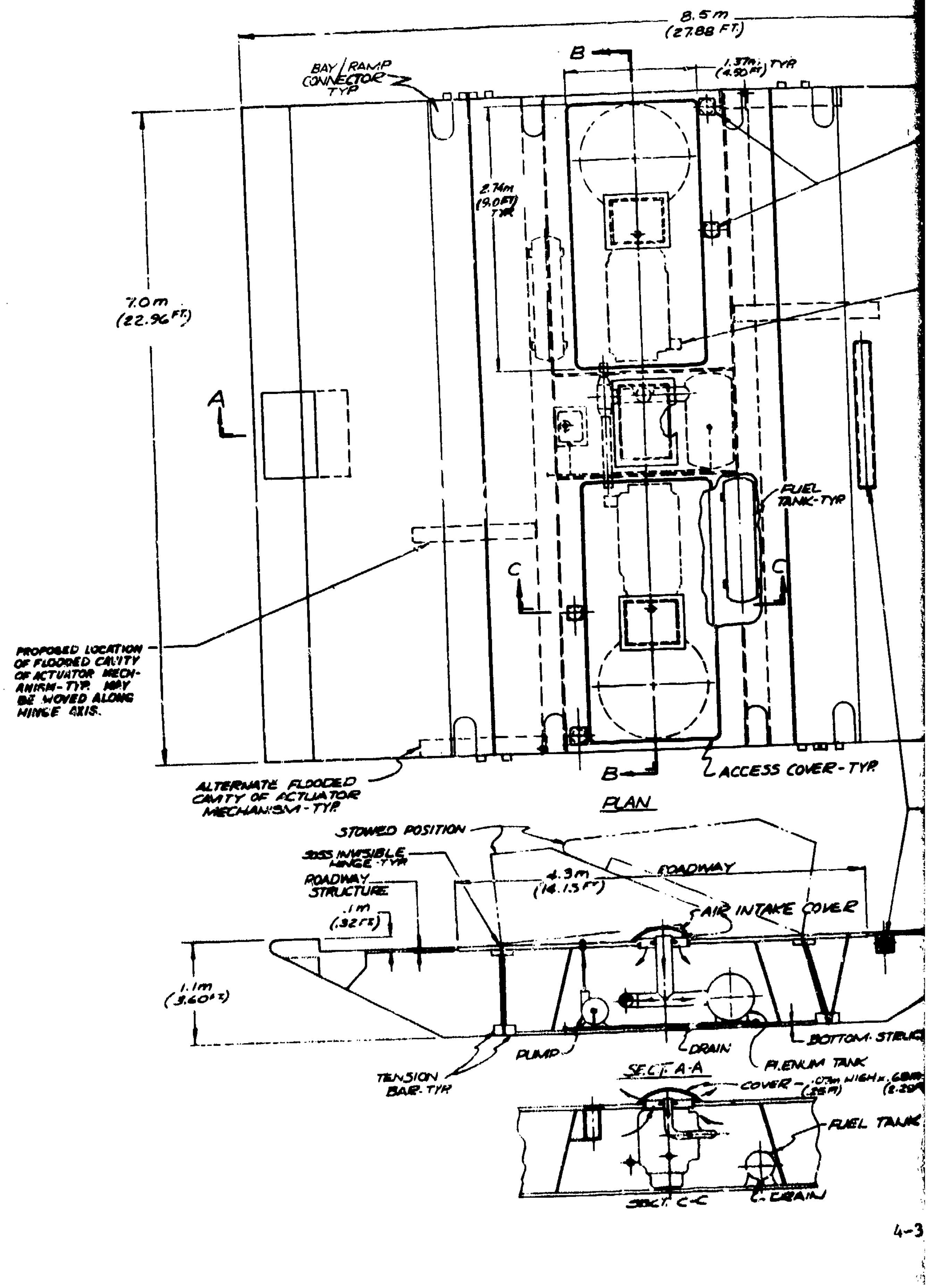
4.1 INTEGRAL PROPULSION BASED ON THE SCHOTTEL PUMP-JET

Figure 4-1 shows an engineered arrangement for a pump-jet propulsion subsystem in a three-ponton bay whose structural design is based principally upon that of the Ribbon Bridge. The width of the bay (and, thus, of the bridge) is 8.5 m; the length of the interior bay is 7.0 m; the depth of the bay is approximately 1 m at the extreme bow areas, where it is 1.1 m. The roadway is 4.3 m wide and extends beyond the center ponton onto each of the bow pontons.

The propulsion compartment is located on the center line lengthwise of the bay. The air inteke for combustion and engine cooling is in the center of the compartment. After passing through a cleaner, the combustion air enters a plenum chamber to remove excessive moisture and then goes into the intake manifolds of the engines. The two engines are Deutz six-cylinder, air-cooled diesels, Model F6L912. Each diesel is clutched to a Schottel Pump-Jet, Model SPJ-32. Each of the pump-jets, which are located at the extreme ends of the bay on the center line, delivers 900 lb of thrust. This design utilizes the smallest Schottel Pump-Jets that are available. The exhaust gases from the engine and the hot air from the cooling of the engines are exhausted on the lengthwise center line of the bay, between the pump-jet and the diesel engine. The openings for intake air and exhaust gas are protected by slightly arched covers.

The control cockpits are midway between the ends of the bay, at the tips of the bow pontons. The propulsion compartment may be entered by moving any one of three sections of roadway. These access covers may be lifted and moved by an auxiliary boom that is stored in the walkway mid-bay. The boom has a chain hoist or other mechanical lifting mechanism suitable for removing and replacing access covers in the roadway; it could also be used to facilitate the connecting of adjacent bays. The two larger covers provide access to the diesel engines and pump-jets. Each of the engine compartments is baffled from the air intake area by a structural member; these and other vertical structural members in the center and bow pontons are designed to enable the roadway to support the heavier military load classes.

Also installed in the center ponton are fuel tanks, a sump pump, and engine-mounted hydraulic pumps that supply the power to open and close the bow pontons. (Opening and closing of the bow pontons is described in section 4.1.2.)


8.5 m
(27.88 FT.)
B
1.37m TYP
(4.50 FT)
BAY/RAMP
CONNECTOR
TYP
2.74m
(9.0 FT)
TYP
7.0 m
(22.96 FT.)
A
FUEL
TANK-TYP
C
C
PROPOSED LOCATION
OF FLOODED CAVITY
OF ACTUATOR MECH-
ANISM-TYP. MAY
BE MOVED ALONG
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B
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STOWED POSITION
SOSS INVISIBLE
HINGE-TYP
ROADWAY
STRUCTURE
4.3m
(14.13 FT)
ROADWAY
.1m
(.32 FT)
AIR INTAKE COVER
1.1m
(3.60 FT)
PUMP
DRAIN
BOTTOM STRUCT
TENSION
BAR-TYP.
PLENUM TANK
SECT. A-A
COVER
FUEL TANK
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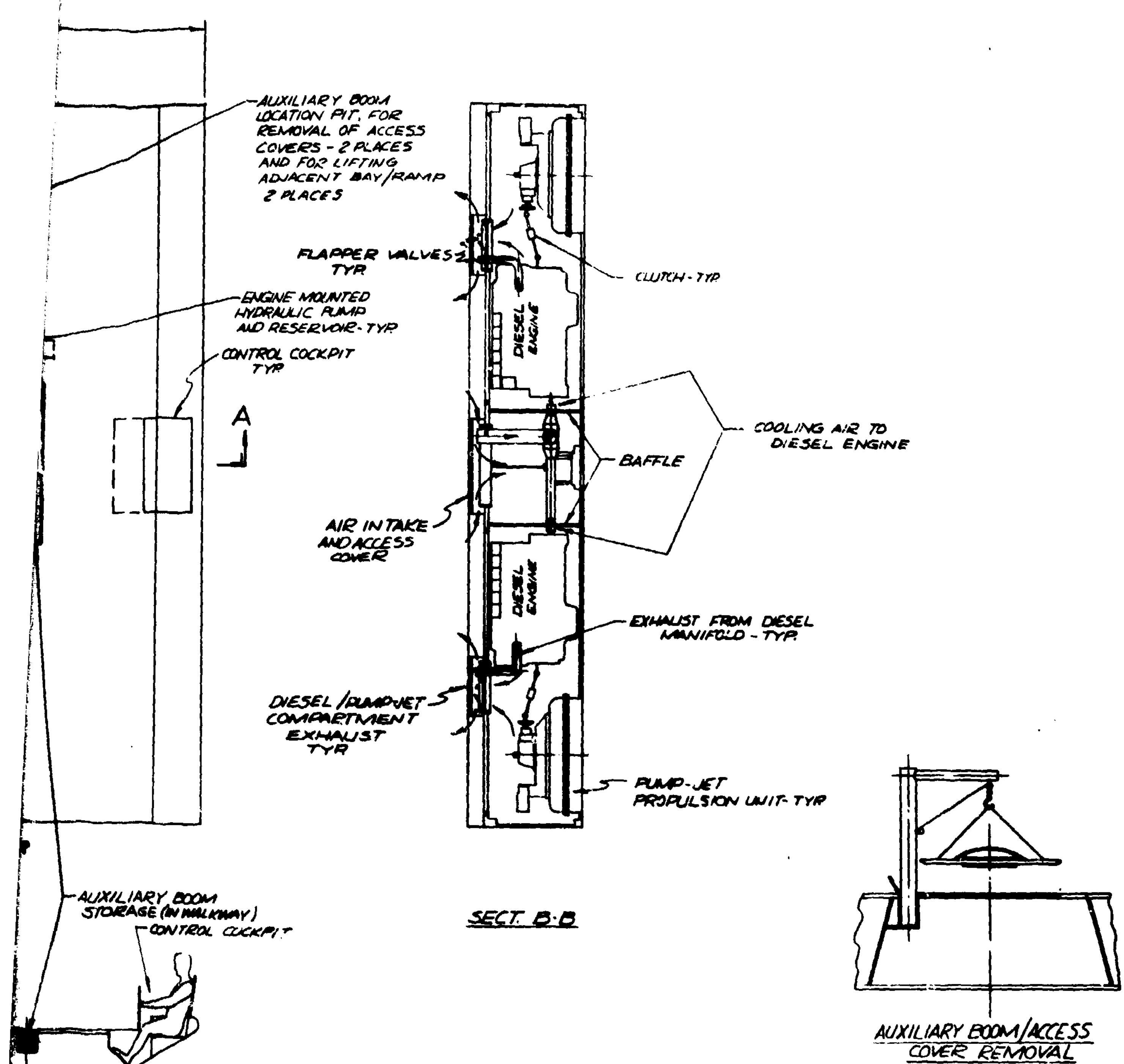
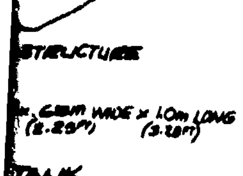
DRAIN





Source: ADL Dwg. SK-61881-4

Figure 4-1

Pump-Jet Propulsion Concept for Three-Part Interior Bay

2.

4.1.1 Capacity, Deadweight, and Buoyancy

Using the same approach as in our structural analysis of the Ribbon Bridge interior bay (see Appendix A), we estimated the characteristics of the three-ponton interior bay. The deadweight of this bay would be approximately 15% greater than that of the interior bay of the Ribbon Bridge. The principal load-bearing structure of the latter bay will be 15% greater than that of the Ribbon Bridge; this fact, together with the calculated volume displacement ratio of the two bays for skin and other supporting mass, indicates that the deadweight of the three-ponton bay would be about 13,800+ lb, or 6.91 ST. The displacement of the bay is estimated at 43.08 ST. Hence, the net buoyancy with the roadway awash would be 36.17 ST. These calculations do not include the deadweight of the propulsion subsystem, which is estimated as follows for one diesel engine and pump-jet:

6 cylinder Deutz diesel	961 lb
Schottel Pump-Jet	243 lb
Pump-jet displacement loss	177 lb
Fuel	130 lb
Clutch, shafting, gearing, controls, supporting structure, fuel tank	400 lb
Total	1911 lb

The deadweight of the dual propulsion subsystem would therefore be 3822 lb, or 1.91 ST. The resultant maximum buoyancy with roadway awash is therefore 34.26 ST.

4.1.2 Hinging and Actuation of Bow Pontons

The roadway extends onto the bow pontons on either side. Our structural engineer initially believed that a continuous, heavy-duty piano hinge would be adequate for the hinging of the bow pontons. However, subsequent discussion and concern for wear from the treads of M1 tanks and other heavy vehicles, convinces us that an exposed piano hinge would be damaged by repeated use, even without abuse. Accordingly, we are now considering use of the invisible Soss hinge, a patented design. The principle of operation of this hinge is shown in Figure 4-2. After discussing the concept and building a miniature model, we believe that this mechanism could be a practical solution to the hinging problem.

The need for an unexposed hinge arises principally from the splitting of the roadway between pontons. Because the center ponton is not sufficiently wide to include the entire roadway, part of the vehicle load must be carried by the bow pontons. However, the bow pontons are not sufficiently buoyant to support this load by themselves, so much of it must be transferred through the hinge to the center ponton. To accomplish this load transfer, the upper mating edges of the pontons must be joined by a series of hinges for the full length of the deck. To preserve the required stowed dimensions, the pivot must be either actually or effectively at the deck surface. A normal hinge so disposed would have its barrel parts above the deck and be subjected to direct contact with the track and wheel debris and treads of the vehicles, which is why the Soss hinge is proposed. Its linkage and arrangement of pins that slide in slots is such as to produce a virtual hinge point on the deck without having any parts exposed when the bow pontons are deployed. The same rugged aluminum roadway extrusions will be used on all three pontons.

Figure 4-3 shows a conceptual design of a mechanism for opening and closing the bow ponton. Power for deployment or stowing is provided by a pair of double-acting hydraulic cylinders connected to the appropriate

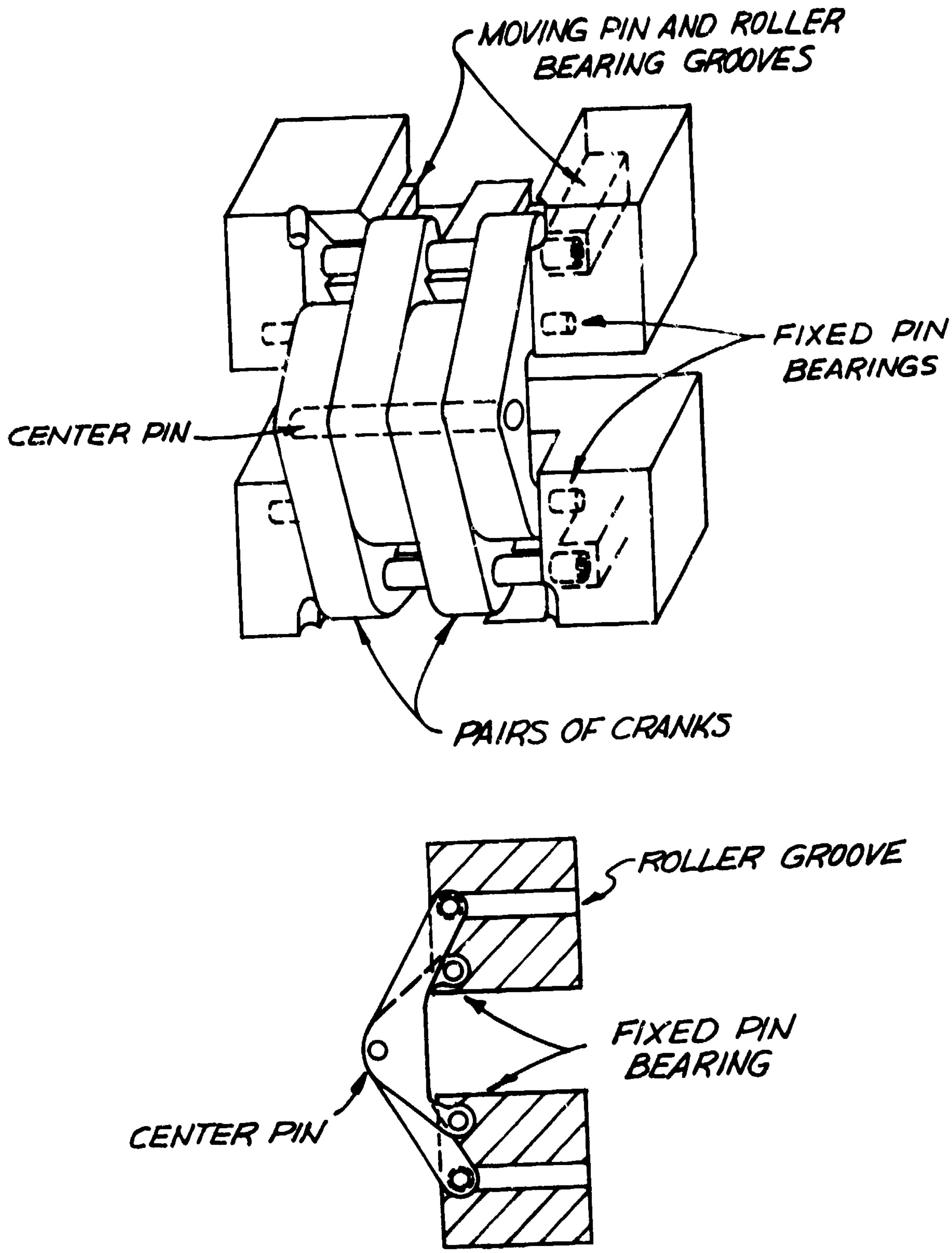
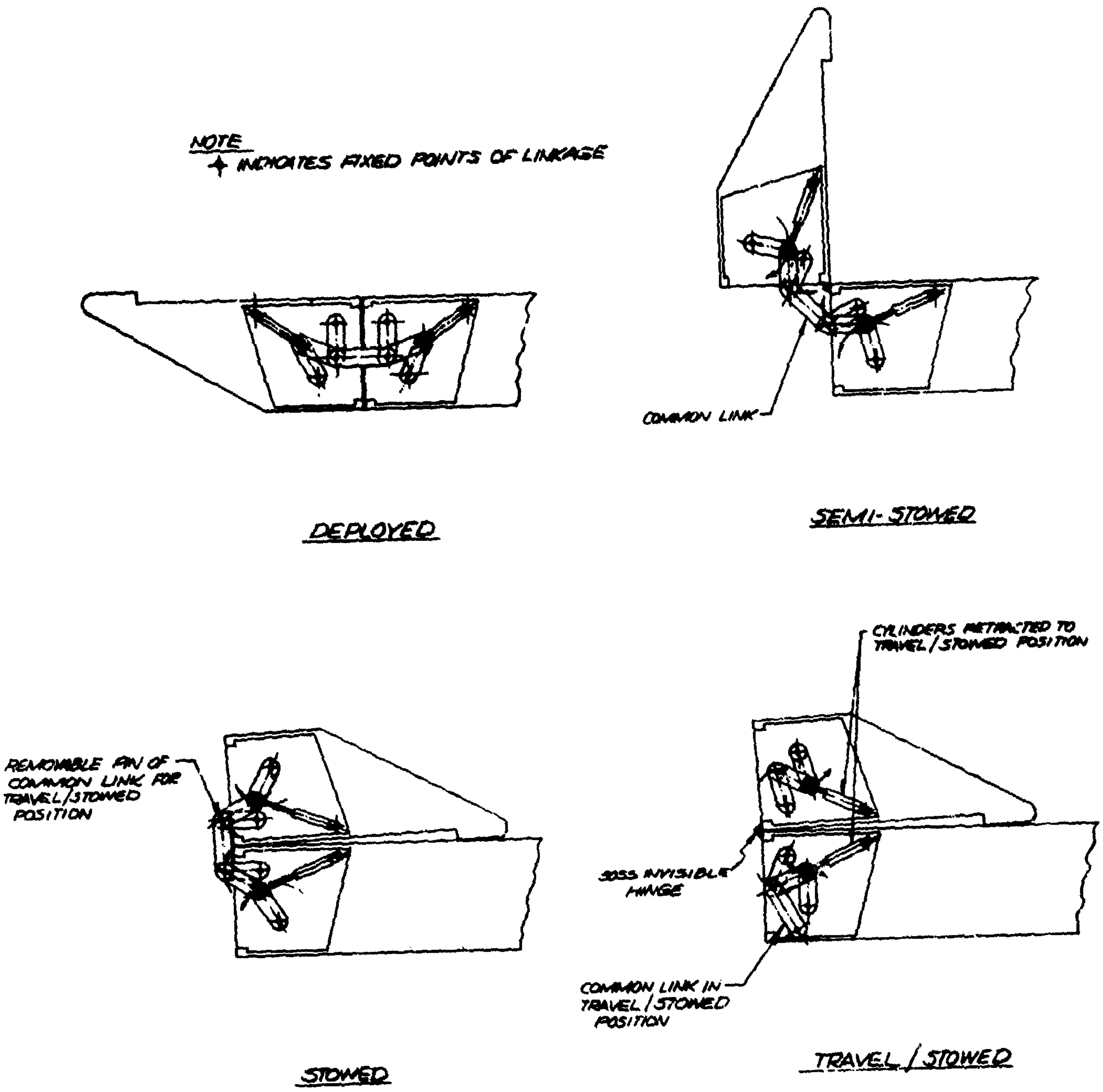


Figure 4-2

Principle of Soss Hinge



Source: ADL Dwg. SK-61881-11

Figure 4-3

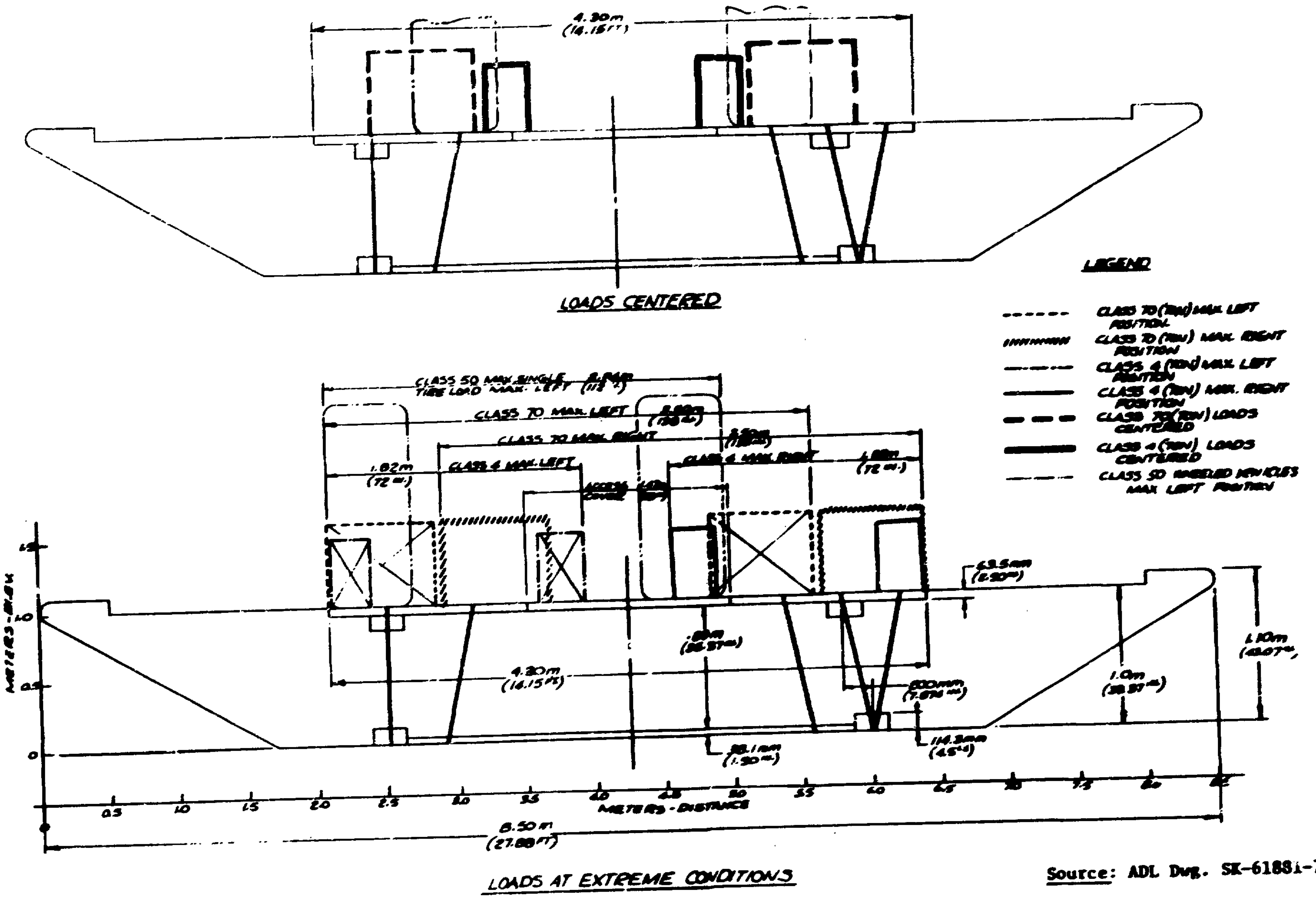
Proposed Actuator Mechanism for Folding Bow Pontons linkages that symmetrically span the separation plane. When the bow ponton is fully deployed, it is locked to the center ponton by a series of hand-operated pins and yokes beneath the hinge at the submerged joint. The connecting arrangement is similar to that between the bays.

To maintain the required stowed dimensions, the pivot of the linkage must be separable after folding, so that the exposed links can be withdrawn hydraulically and stored within the envelope of the folded pontons. Further study should produce a linkage that remains totally within the allowable envelope without need for disconnection when stowed.

4.1.3 Roadway Width and Effect of 70-Ton Loads

Figure 4-4 shows the location of Military Load Classes 4 and 70 on the three-ponton interior bay when centered and when at extreme position; Load Class 50 is shown for wheeled vehicles at maximum left position only. The roadway was increased in width to 4.3 m as a ratio of vehicle width of MLC 70 over MLC 60, based on the width of the Ribbon Bridge roadway as acceptable for MLC 60. The roadway extends into the bow ponton on the left by approximately 600 m. Figures 4-1 and 4-4 show the structure which contains the propulsion compartment and the additional structure in the right bow ponton.

Four main tensile-load-carrying steel bars are provided--two in the center ponton and one in each bow ponton--all located at the bottom adjacent to the separation planes. The pins and yokes that secure the bow pontons to the main roadway ponton are attached to these bars laterally along the length of the bay. At the ends of the bars are the bay-to-bay pins and yokes, which are connected in the same manner as in the existing Ribbon Bridge. To provide a tensile capability for both the main roadway



Source: ADL Dwg. SK-61881-7

Figure 4-4

Position of Military Load Classes 4 and 70 on Three-Part Interior Bay ponton and the bow pontons, there are yokes on all four bars. Their position and number (one male and one female per end) are such as to allow interior bays and ramp bays to be attached to either end of another interior bay. Roadway connectors are the same as in the Ribbon Bridge.

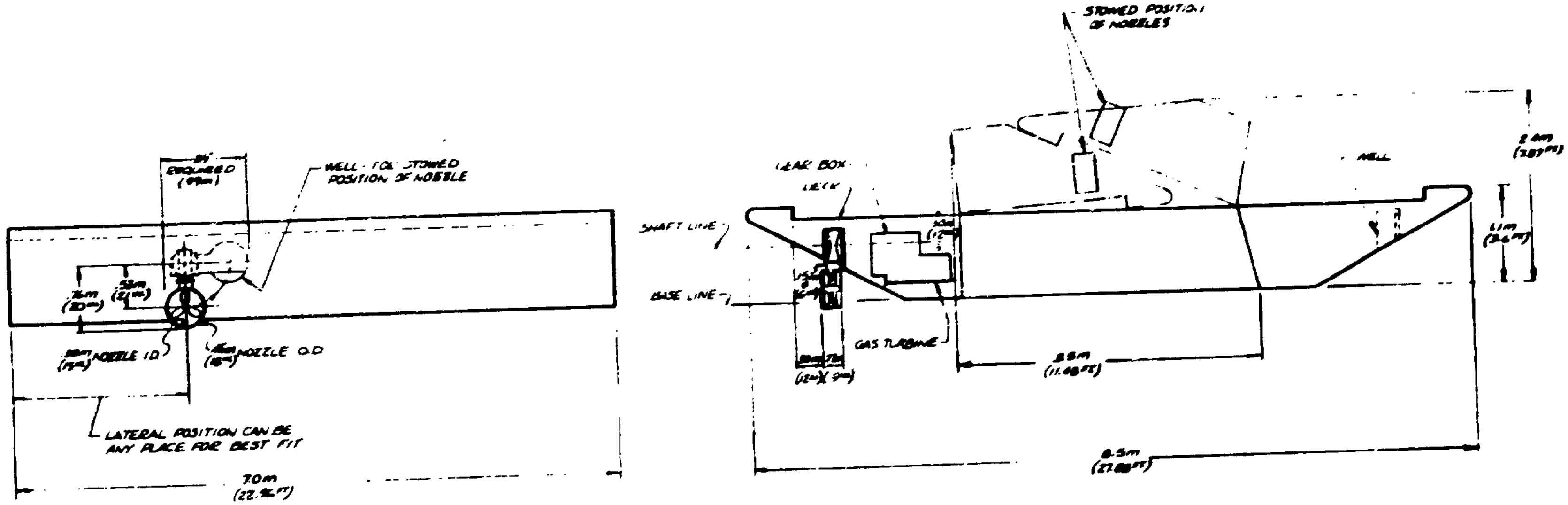
4.2 OUTBOARD DRIVE CONCEPT FOR THREE-PONTON INTERIOR BAY

In the outboard drive concept (Figure 4-5), a Schottel Model SPR-12 outboard drive is driven by a Turbomach gas turbine, Model T-62T-32. The outboard drive rotates and is stowed in a well in the bow ponton for protection during launch. Following launch it is rotated 90°. In operating position, it extends approximately 3 in. below the base line or bottom of the bay. The lateral position of the outboard drives in the bow pontons would be balanced for symmetry and effective maneuverability; their exact location would depend on the details of the structure.

We attempted to find a small, variable-pitch propeller to avoid a variable-speed drive between the gas turbine and the outboard drive, but no standard propeller of this kind is made in the size needed; a special procurement would be required. The gas turbine was chosen as the prime mover because, as discussed earlier, it can withstand being stowed upside down or at 180° from the operating or unfolded position.

We do not recommend the outboard concept as a means of adding integral propulsion to the three-ponton bay. Its initial cost is high because of the gas turbine prime mover, and the requirement for a variable-speed drive or propeller adds complexity. Moreover, an outboard drive has roughly twice the fuel consumption of a pump-jet.

The displacement volume of the three-ponton bay with outboard propulsion may be calculated by the following formula:



Source: ADL Dwg. SK-61881-3

Figure 4-5

Outboard Drive Concept for Three-Ponton Interior Bay

$$V = 37.50\ h + 3.46\ h^2$$

where:

V = displacement volume (m^3)

h = displacement height or draft (m)

Or, if V is known and h is not, the following formula may be used:

$$h = \frac{-10.32 \pm \sqrt{10.32^2 + 1.16V}}{2}$$

From the former equation, the maximum displacement of the three-part interior bay is 39.16 m^3 in fresh water (density = 1.10 ST/m^3). The maximum displaced weight would be 43.08 ST.

The deadweight of the propulsion subsystem for the outboard drive is estimated as follows:

Gas turbine	85 lb
Variable-speed transmission	1,000 lb
Outboard drive	617 lb
Fuel (100 lb/hr, 5 hr)	500 lb
Fuel tank, piping, structure	300 lb
Displacement loss	152 lb
Total	2,654 lb or 1.33 ST

4.3 INTEGRAL PUMP-JET PROPULSION FOR THREE-PONTON RAMP BAY

As mentioned at the beginning of section 4, height restrictions in the ramp bay complicate the installation of a pump-jet propulsion subsystem. To facilitate the conceptual design, it was necessary to develop a scale model of the three-ponton ramp bay with all of its essential functional attributes. Figures 4-6 through 4-9, which are photographs of the model, show various features of the ramp bay in folded and unfolded modes.

Figure 4-6

Three-Part Ramp Bay Unfolded in Operational Position at Bank with Approach Ramp Lowered

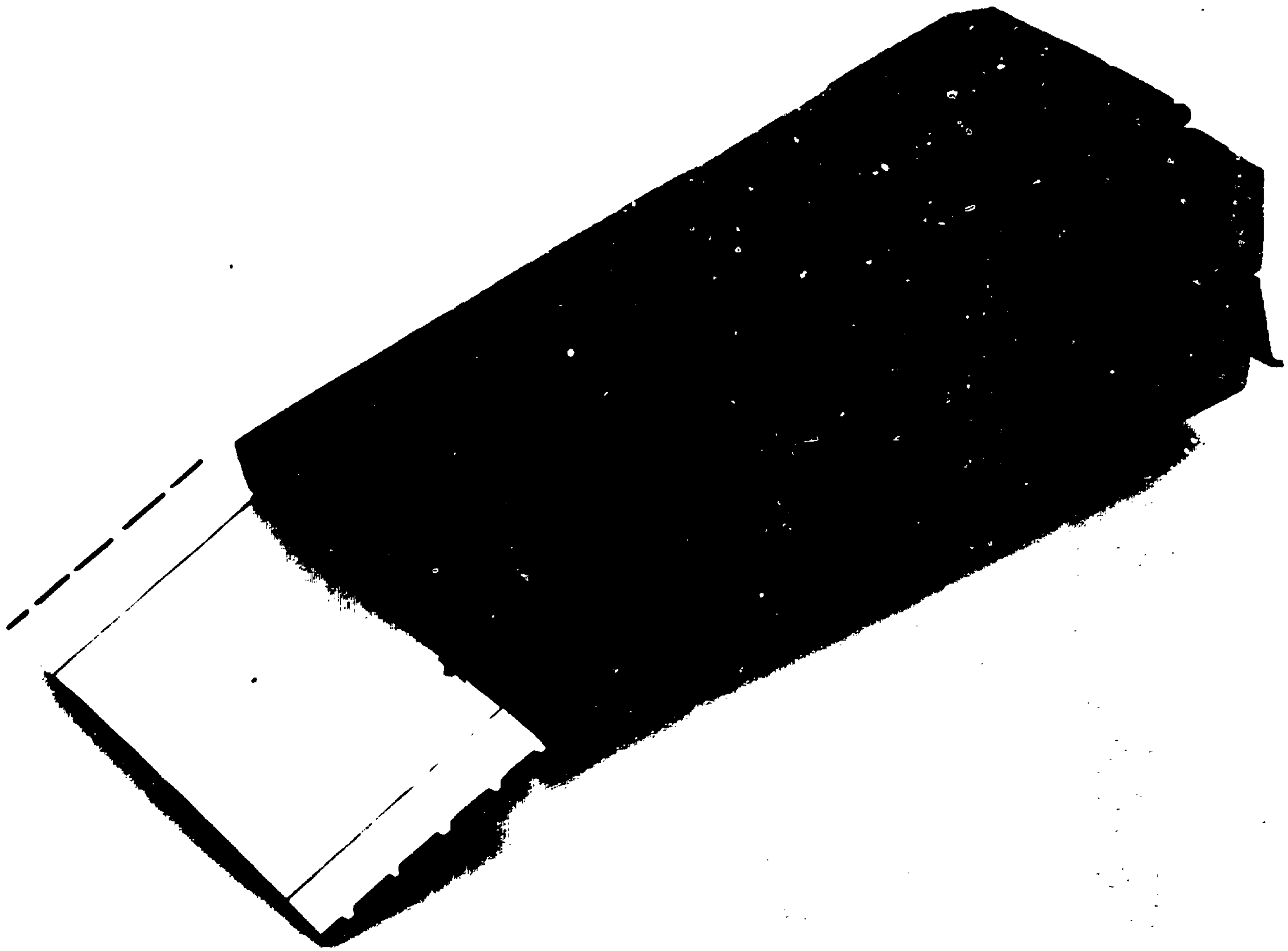


Figure 4-7

Three-Part Ramp Bay Folded in Stowed Position with Approach Ramp Lowered, Showing Cantilevering of Upper Bow Ponton

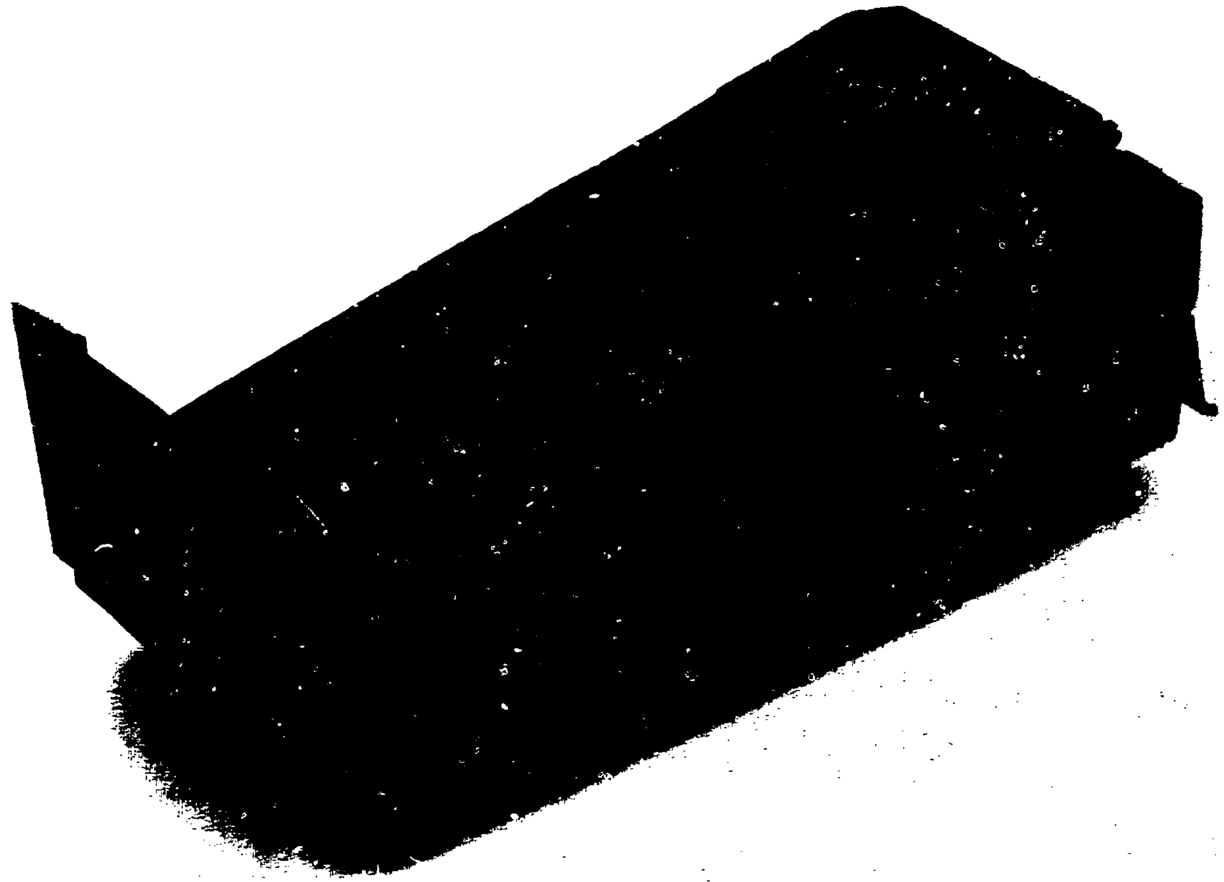


Figure 4-8

Ramp Bay in Folded Secured Position with Approach Ramp Folded and Secured to Upper Bow Ponton

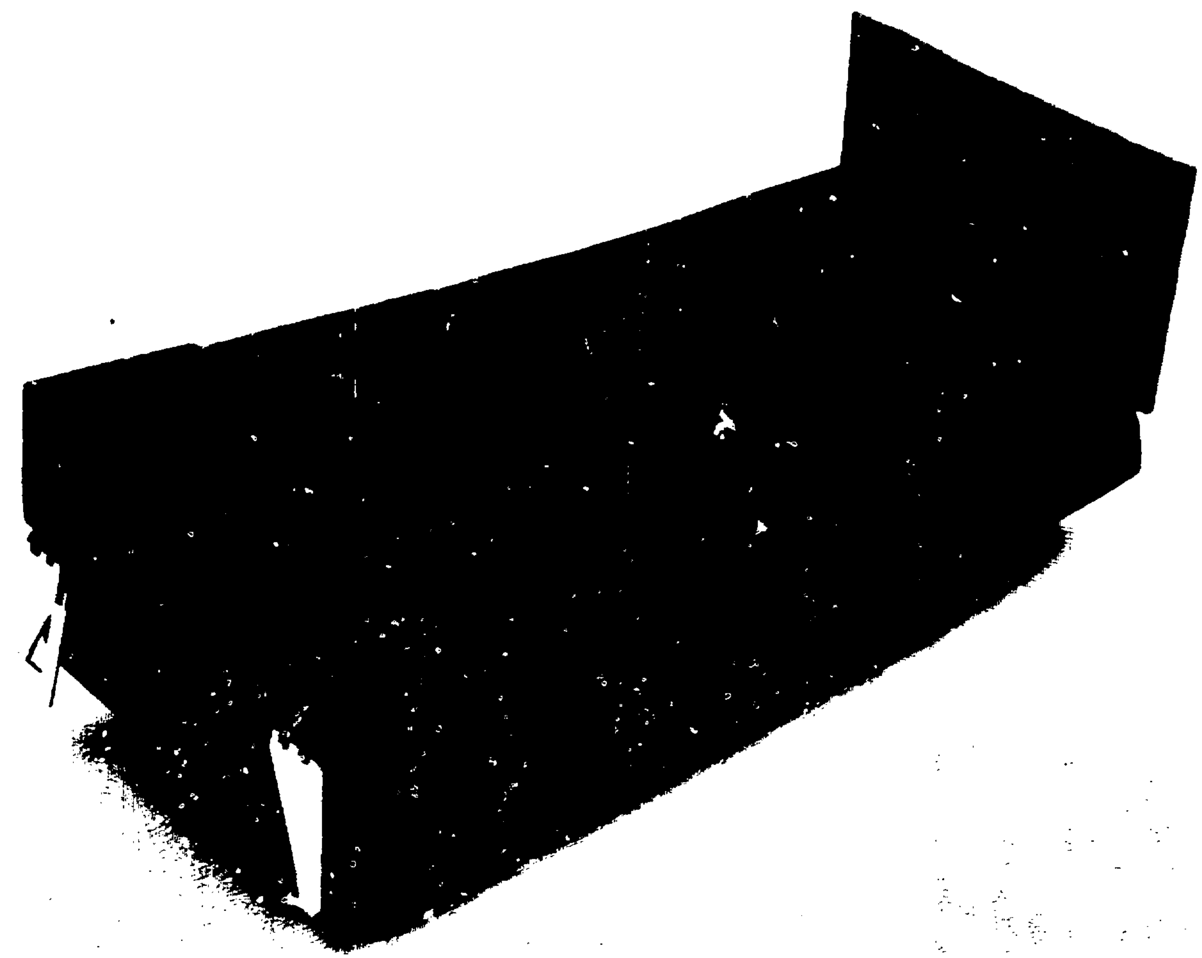


Figure 4-9

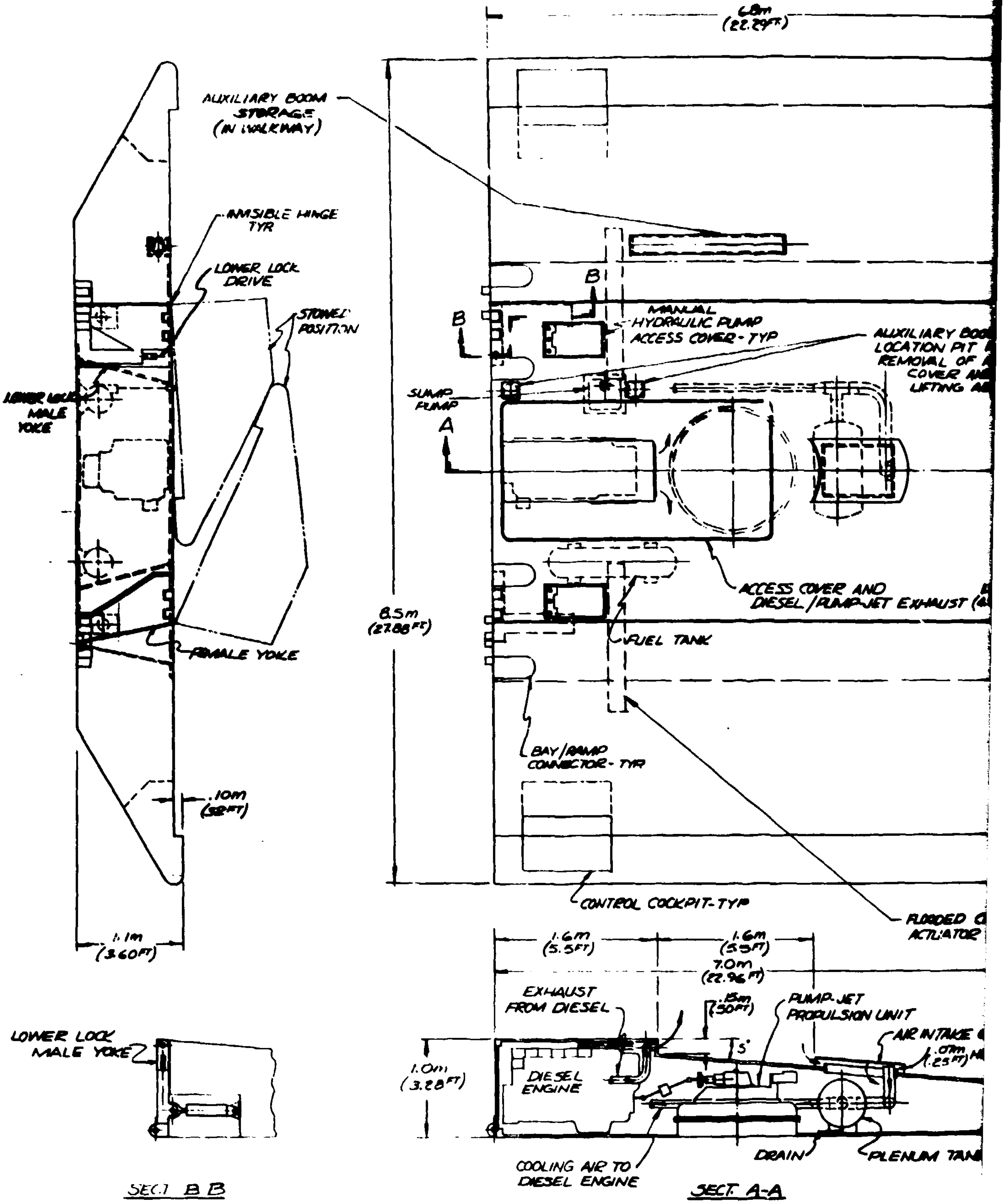
Ramp Bay Folded and Secured, Showing Interior End with Male and Female Ramp Elevation Yokes

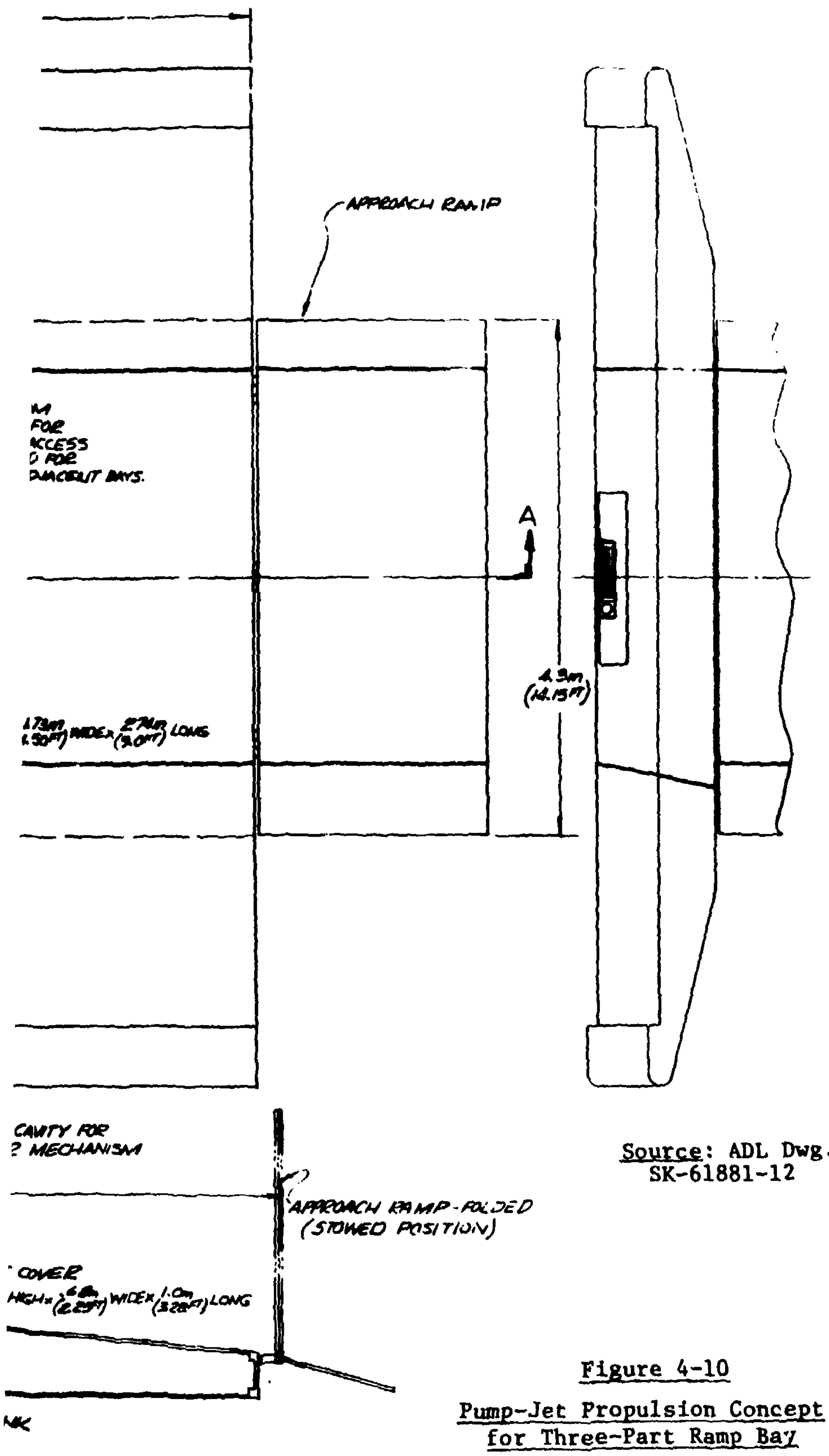
The pump-jet propulsion concept for the ramp bay is presented in Figure 4-10. To provide sufficient height for the six-cylinder diesel engine within the roadway ponton, it is necessary to raise an area approximately 1.6 m long and 0.6 m wide in the middle of the roadway at the top of the ramp. Although the Schottel Pump-Jet fits within the present envelope of the roadway ponton, another protrusion in the roadway is needed for the air intake. The outlet cooling air and diesel exhaust can be expelled from the housing over the diesel. Since the roadway extends onto the two bow pontons, a three-part approach ramp is needed that is foldable in two planes.

The general construction features of the ramp bay are the same as those of the interior bay described above. In addition, hydraulically operated male and female ramp elevation yokes are provided; these connect to the roadway with double pins and links (hinge-like) and to the tensile yokes at the bottom in the same manner as an interior bay. There is adequate room at the interior end of the ramp for both the elevation cylinders (which can hold the entire ramp at an angle to the interior bay) and the folding cyliners. (See Figure 4-10.)

The approach ramp is composed of three hinged sections which, when folded and stowed, sit vertically. Latches on the approach ramp secure the shore end of the folded bow pontons for transport. When the approach ramp is stowed, its free end is at the same height as the interior end of the folded bow ponton. (See Figure 4-8.)

The deadweight of the structure for the three-ponton ramp bay is estimated at 13,455 lb, or 15% greater than that of the Ribbon Bridge ramp bay. The deadweight of the single Schottel Pump-Jet propulsion subsystem


AUXILIARY BOOM STORAGE (IN WALKWAY)
INVISIBLE HINGE TYP
LOWER LOCK DRIVE
STOWED POSITION
LOWER LOCK MALE YOKE
FEMALE YOKE
.10m (.33FT)
1.1m (3.60FT)
6.8m (22.29FT)
B
MANUAL HYDRAULIC PUMP ACCESS COVER - TYP
SUMP PUMP
A
ACCESS COVER AND DIESEL / PUMP-JET EXHAUST
FUEL TANK
8.5m (27.88FT)
BAY/RAMP CONNECTOR - TYP
CONTROL COCKPIT-TYP
FLOODED
ACTUATOR
LOWER LOCK MALE YOKE
SECT B-B
1.6m (5.5FT)
1.6m (5.5FT)
7.0m (22.96FT)
EXHAUST FROM DIESEL
PUMP-JET PROPULSION UNIT
AIR INTAKE
1.0m (3.28FT)
DIESEL ENGINE
COOLING AIR TO DIESEL ENGINE
DRAIN
PLENUM TANK
SECT. A-A



Source: ADL Dwg.
SK-61881-12

Figure 4-10

Pump-Jet Propulsion Concept
for Three-Part Ramp Bay

is estimated at 2021 lb. Hence, the deadweight of the ramp bay and its propulsion subsystem is 7.74 ST. The net buoyancy of the ramp bay when attached to an interior bay at a 15° angle is 5.48 ST with the interior roadway awash. The ramp bay net buoyancy vs. draft of the interior bay with the ramp bay at 15° is plotted in Figure 4-11. (The related calculations are presented in Appendix A.)

4.4 TRANSPORTER FOR IMPROVED WET BRIDGE

The deadweight of the three-ponton interior bays, including the pump-jet propulsion system, is 8.94 ST. The deadweight of the corresponding ramp bay is estimated to be 7.74 ST. To assure off-the-road mobility, the transporter should have a 10-ton capacity. Contact was made with the Commander, USATACOM, Attention: Dennis Mazurek, DRSTA-RTE, Warren, Michigan 48090, to determine the availability of the Heavy Expanded Mobility Tactical Truck (HEMTT). Data from Oshkosh Truck Corporation were provided (see Appendix B); these were used as the basis for the transporter concept for the Improved Wet Bridge. Drawings of the HEMTT carrying the two types of bays are shown in Figure 4-12. The transporter would require a modified bed similar to that of the Transporter, Floating Bridge used to carry the Ribbon Bridge bays.

A launch and retrieval subsystem (including roller assemblies, winch cables, and a boom) would be needed. It would have to be designed and engineered so that the roller subsystem is at the chassis level, to assure that the stowed bays are well within the acceptable envelope. Both the interior and ramp bays fit within the required envelope when stowed on the transporter; in fact, the height of the ramp bay is slightly less than the 3.9-m limit.

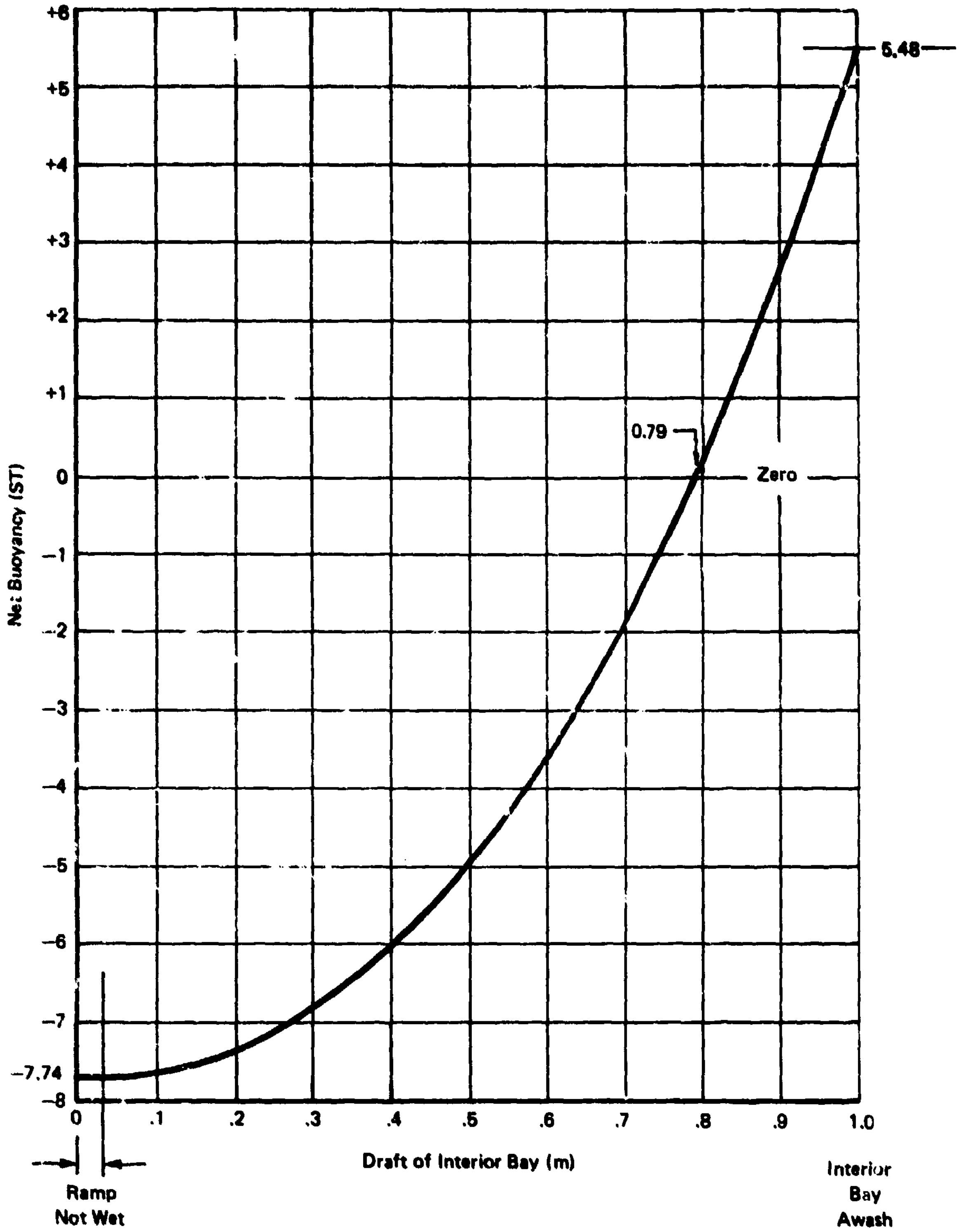
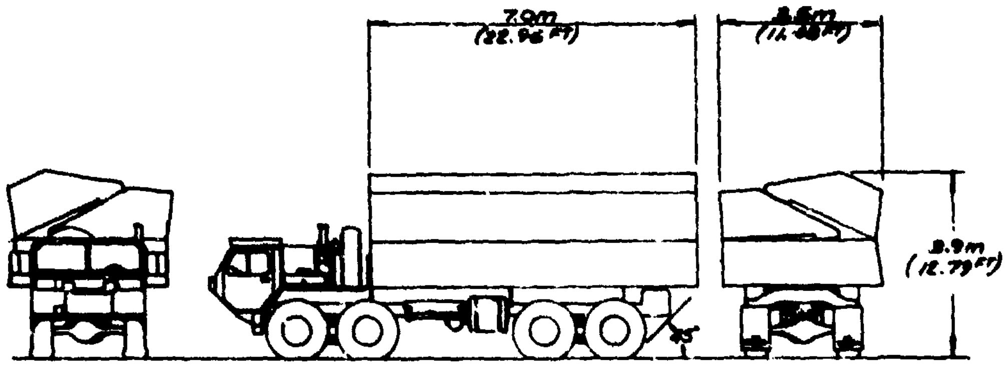
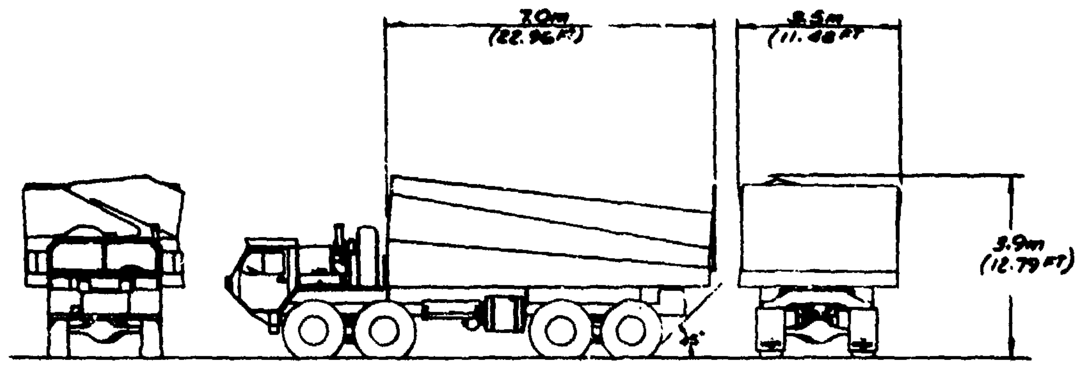


Figure 4-11

Ramp Bay Net Buoyancy vs Draft of Interior Bay with Ramp at 15°



Transporter with Interior Bay



Transporter with Ramp Bay

<u>Source</u>: ADL Dwg. SK-61881-9

<u>Figure 4-12</u>

<u>HEMTT Transporter for Three-Part Improved Wet Bridge Bays</u>

4.5 STILLWATER RAFTING DRAFT AND FREEBOARD

Using the capacity calculations and curves for the Ribbon Bridge (Current and Improved) and the Improved Wet Bridge, we calculated the draft and freeboard for various loads on three-bay, four-bay, and five-bay rafts. Ramp bays would be at a 15° angle to the interior bays for each of the three sizes of rafts.

Figure 4-13 applies to the Ribbon Bridge. It should be remembered that for the Improved Ribbon Bridge, two half-bays comprise a single bay on the graph. Thus, a three-bay raft would consist of two integral propulsion half-bays and two ramp bays. A four-bay raft could consist of four integral propulsion half-bays and two ramp bays, or one Ribbon Bridge interior bay, two integral propulsion half-bays, and two ramp bays. The five-bay raft could consist of two Current Ribbon Bridge interior bays, two integral propulsion half-bays, and two ramp bays, or one Current Ribbon Bridge interior bay, four integral propulsion half-bays, and two ramp bays. It is highly unlikely that all interior Current Ribbon Bridge bays would be replaced by pairs of integral propulsion half-bays.

Similar plots of load vs. freeboard are shown in Figure 4-14 for the Improved Wet Bridge. Table 4-1 summarizes data from Figures 4-13 and 4-14 to show the extent to which Improved Wet Bridge rafts excel Ribbon Bridge rafts at two values of roadway freeboard. The Improved Wet Bridge with integral propulsion three-ponton bays can carry 10 tons more on a three-bay raft and at least 20 tons more on a four- or five-bay raft.

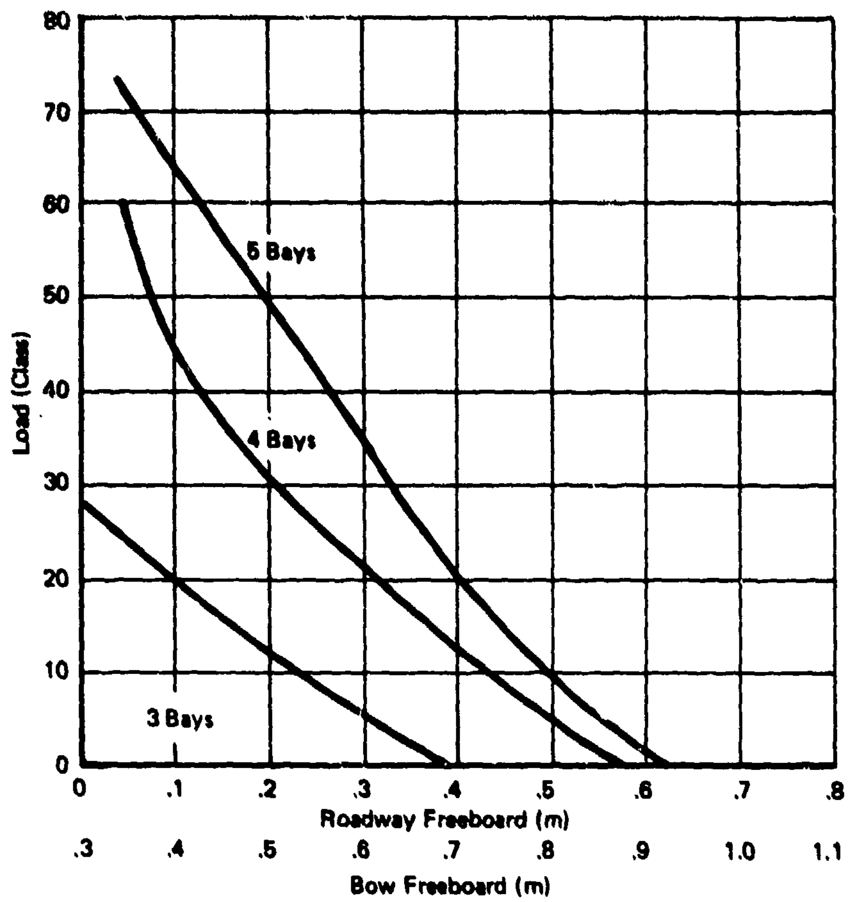


Figure 4-13

Stillwater Rafting Freeboard vs Load:
Current or Improved Ribbon Bridge, Ramp Angle 15°

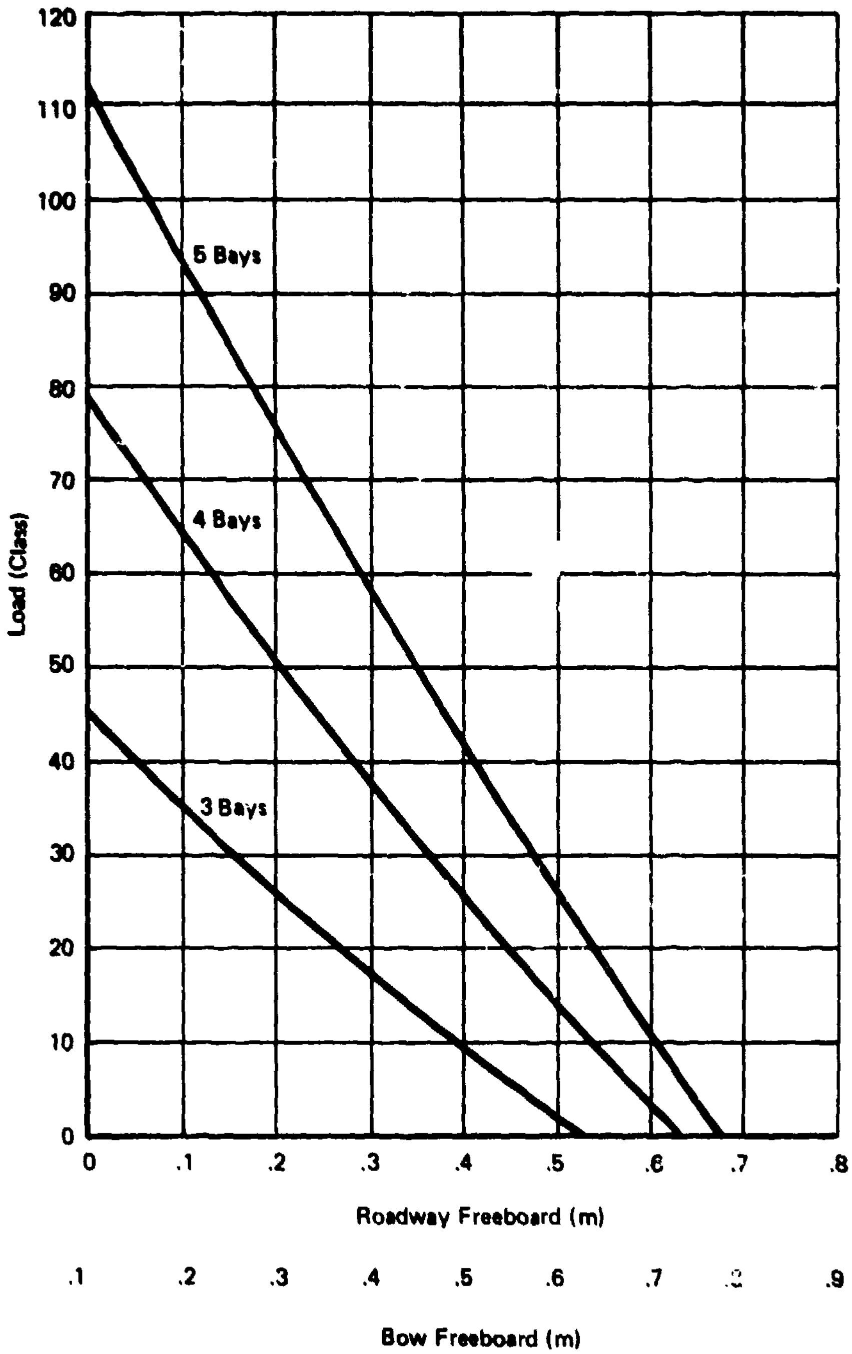


Figure 4-14

Stillwater Rafting Freeboard vs Load: Improved Wet Bridge, Ramp Angle 15°

Table 4-1

Raft Capacity of Alternative Bridge Systems for Stillwater at 0.1 m and 0.2 m Roadway Freeboard

No. of Bays	Roadway Freeboard	Military Load Class: Current or Improved Ribbon Bridge	Military Load Class: Improved Wet Bridge
3	0.1	20	30+
	0.2	10+	20+
4	0.1	40+	60+
	0.2	30+	50+
5	0.1	60+	90+
	0.2	50	70+

In Figure 4-13 note that the freeboard for the bow pontons is approximately 0.3 m greater than that of the roadway; in Figure 4-14, the difference is only 0.1 m. This difference in freeboard reflects the design of the two systems. A comparison of bow freeboards for the same load class, however, indicates a greater similarity between the Ribbon Bridge and the Improved Wet Bridge. For instance, from Figure 4-13, for a 70 Military Load Class the bow freeboard would be 0.36 m for a Current or Improved Ribbon Bridge raft. With the same load on a five-bay raft of the Improved Wet Bridge system, the bow freeboard would be 0.34 m.

5. CONTROL SYSTEM

5.1 GENERAL PRINCIPLES

We assume that each separate unit (whether full-bay or half-bay) is fitted with two steerable propulsors. Since the units are double-ended, they require a control station at each end. Each of these duplicate stations requires the following:

2 power controls

2 directional controls

Start/stop switch, fuel controls, and instrumentation

To facilitate one-handed operation, various types of dual-motion controls could be used, such as a rotational motion for direction and an axial or angular motion for power. Two such controls, one for each propulsor, would provide the basic operational control for the unit. Because the control stations are not adjacent to the propulsor, and also because remote control is needed, an electro-hydraulic system is indicated.

When some or all of the units are self-propelled, remote control is desirable to reduce manpower needs and exposure. On the other hand, we assume that no more than perhaps four units can be effectively supervised by one operator. Thus, the following discussion applies to a maximum of eight individual power plants.

The first simplification lies in ganging rows or columns of propulsors when the units are assembled into a bridge or large ferry. Further examination shows that the simplest approach lies in ganging only the two propulsors of each unit. The reason for this is as follows: With all units self-propelled, the span between propulsors in one unit may be 9 ft, whereas the distance between propulsors in adjacent units will be 22 ft. (If the propulsors are in the bow pontons, the span will be abcut 22 ft. also.) When integral propulsion units are alternated with non-powered ones, the distance between propulsors in each unit may be about 15 ft. while the distance between adjacent propulsors will be about 32 ft. Therefore, since the propulsors in the two separate units can exert at least as large (possibly twice as large) a turning moment as the propulsors in a single unit, it is preferable to control adjacent units separately and gang the forward and aft propulsors. In longitudinal rafting, with the powered units lashed alongside, forward-and-aft ganging is desirable when multiple powered units are used or when they are not individually manned.

In the proposed system, the controls in each unit would be ganged by an electrical switch in the unit, and the combined controls would be connected to the control point by a cable that is stored in each unit on a reel. The central control would be a portable console that could be installed in any of the control cockpits. It would contain receptacles for each of the cables from the controlled units and would obtain any necessary electrical power from the unit on which it was installed.

5.2 ALTERNATIVES

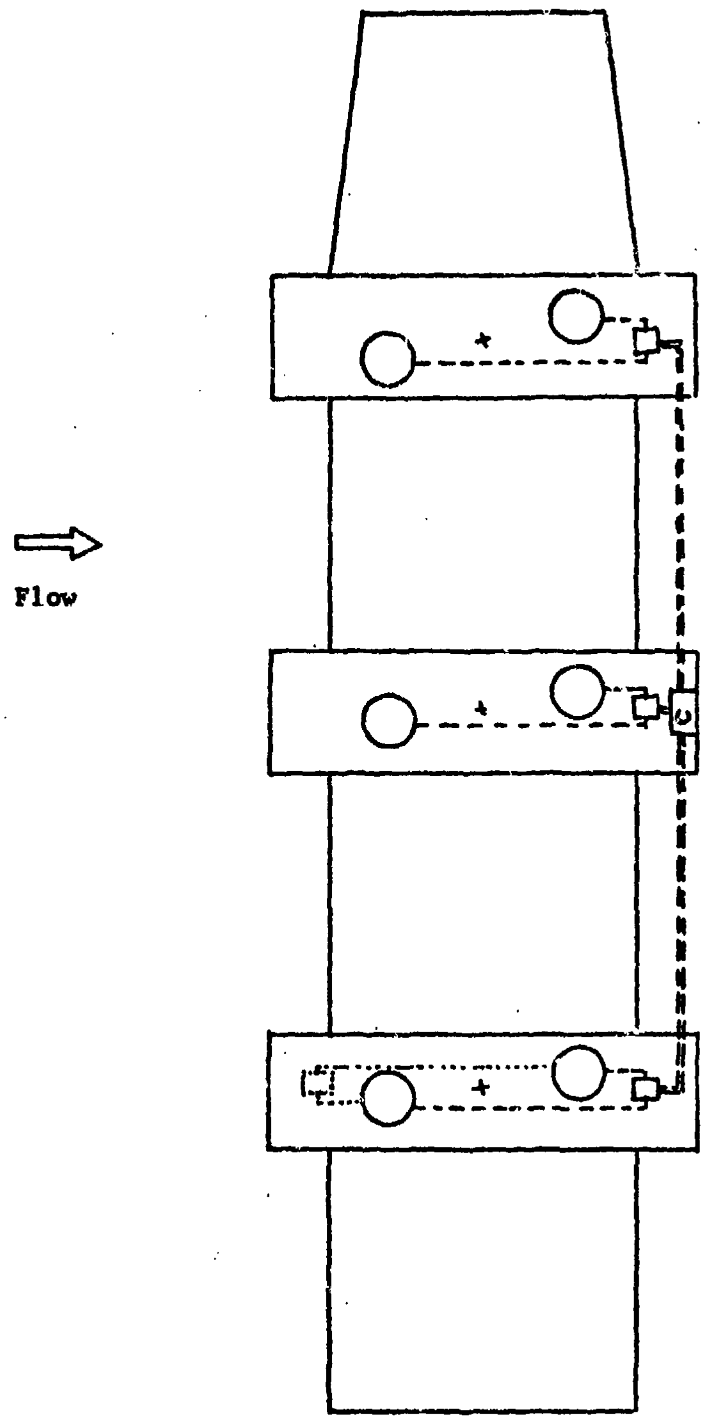
The console could be designed in either of two ways:

(1) It could merely centralize the four sets of power and direction controls without ganging them; it would also incorporate such items as an alarm for each plant and an emergency cut-off for each plant. Adjustment of the multiple power and direction settings to achieve the desired results would be the responsibility of the operator, who would need training and experience.

(2) Alternatively, the console could incorporate a computer that would separate the desired force, direction, and moment into the several components that can be provided by the individual units and automatically transmit the proper control signals to them. Resolution of the component forces would require the solving of three reasonably simple simultaneous equations, but the reverse process would entail additional assumptions or ground rules concerning the number of units and the intervals between them, which would also have to be fed into the computer.

Selection between these two alternatives depends on the relative importance of console size, cost, and complexity and the requirements for ope tor skill and training. In bridging operations, control is fairly simple because it involves steady, or only slowly changing, conditions. During ferrying operations, however, the requirements for maneuvering would make the more automated system desirable. The powered units would probably be individually manned during bridge assembly, so remote operation would not be of concern at that time.

Figure 5-1 shows a raft or section of bridge with three powered units alternating with non-powered units. The propulsors are indicated by small rectangles. The control arrangement would be identical if all units were self-propelled, or if there were four powered units, or if there were more non-powered units between the powered ones. The proportions of the non-powered units are those of the standard interior bays; the proposed powered units are approximately half-bays.



Scale: 1 in. = 16 ft

Figure 5-1

Schematic Diagram of Control System for Raft with Three Powered Bays

When the two propulsors of a bay or unit are ganged and are producing approximately the same thrust in the same direction, they impose no turning moment. Thus, in adding the effects of multiple units, the force exerted by each unit may be taken as a simple force equal to the total thrust of its two propulsors and exerted at the center of the unit, as marked on the diagram.

In the powered unit at the lower part of Figure 5-1, an alternate control path to the other end of the unit is shown. The units must be operable from either end, and it would also be desirable to have redundancy in the event of damage to one control station. Cables from the two propulsors lead to the unit's control station, indicated by a small box, where they are ganged. At this point provision would be made for small local adjustments and synchronization as needed to improve performance of the unit and to adjust for variations in water velocity along the length of the bridge.

The more comprehensive control system is indicated by the broken double line, which leads to the control station (c) situated, in this case, on the center powered unit. This station is the portable console which is installed at a convenient location for supervising the operation, or at least for supervising one set of bays or units. The array shown in Figure 5-1 is equivalent to a five-and-one-half bay raft. A ramp ponton is shown at one end, but ramps could be used at both ends.

To summarize our recommendations:

- The propulsors of each unit--whether it be a full bay or half-bay--should be ganged in both power and direction. This permits the unit to be treated as one thrust vector by the control system.

- Fine adjustments for power level and synchronization of the two propulsors on each unit should be performed locally from each unit's own control station.
- The combined control for an assembly of up to four powered units should be incorporated in a portable console that can be placed in any of the control cockpits.
- The console should be linked by electric cables to each powered unit. A cable should be carried on a reel in each unit for this purpose.
- The extent of the combined control should be investigated further, since individual manual settings for each unit may be preferable to automatic computed development of the combination of vectors needed to achieve a desired moment and thrust.
- The integrated control system should have no automatic feedback loop; the only feedback should be provided by the operator via visual references or directional sensors.

5.3 MULTIPLE PROPULSION UNITS

This section briefly considers the mathematical aspects of computer control of a multi-unit array. The following assumptions apply:

(1) Each powered unit in the array has two steerable propulsors.

(2) The two propulsors in each unit are ganged in both force and direction, so that the resultant provided by each unit acts as a single vector (F_1 θ_1) with origin halfway between the two propulsors.

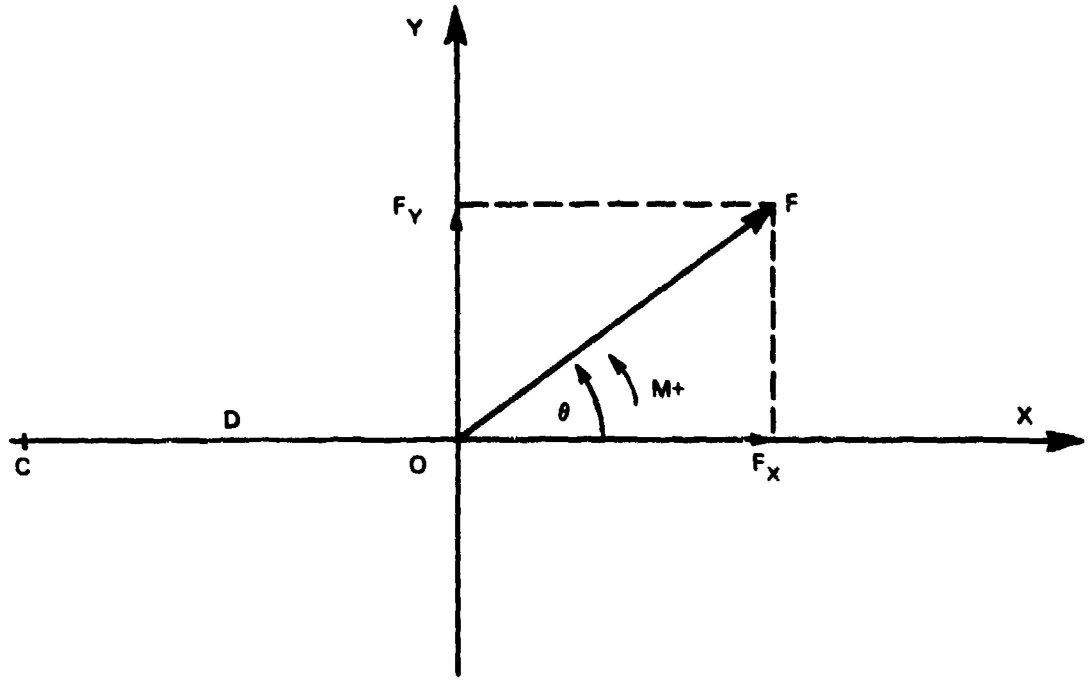
(3) Each unit is a known distance from the estimated center of resistance of the array of units ($\overline{OA} = a_1$).

Figure 5-2 shows the standard derivation of the thrust, angle, and moment which result, and the summation for N powered units. Since there are three simultaneous governing equations and 2N variables, an automatic integrating controller requires additional conditions in order to select individual sets of values of F_i and θ_i.

The formulation of these conditions is complicated by the fact that the control unit must be adaptable for two-, three- or four-unit arrays and that the conditions set may differ from each size array. The central controller would have to be programmed to recognize certain required levels of thrust and certain proportions of thrust to moment and to adapt power and direction allocations accordingly.

Figures 5-3 and 5-4 indicate the type of approaches which can be applied to produce soluble relationships. In Figure 5-3, the angles of the three propulsion units are kept equal, thereby eliminating three of the six variables; the three force (or thrust) vector magnitudes remain, together with the three equations. The envelope of achievable, simultaneous moments and forces--in terms of the maximum force of one unit and of unit spacing--is shown. If the equation for F_2 is relaxed, the envelope can be extended into the region labeled II.

Figure 5-4 shows an alternative in which the three units are set at equal power but at different angles. This again eliminates three variables. The solution of the equations in this example is appreciably more complex. By inspection, it is evident that the maximum values of moment and thrust are more limited than in the first case. Unless other factors enter into the picture, the first approach would be simpler and more effective except for longitudinal rafting.



$F_X = F \cos\theta$

$F_Y = F \sin\theta$

About Point C:

$M_X = 0$

$M_Y = F_Y \overline{CO} = D F \sin\theta$

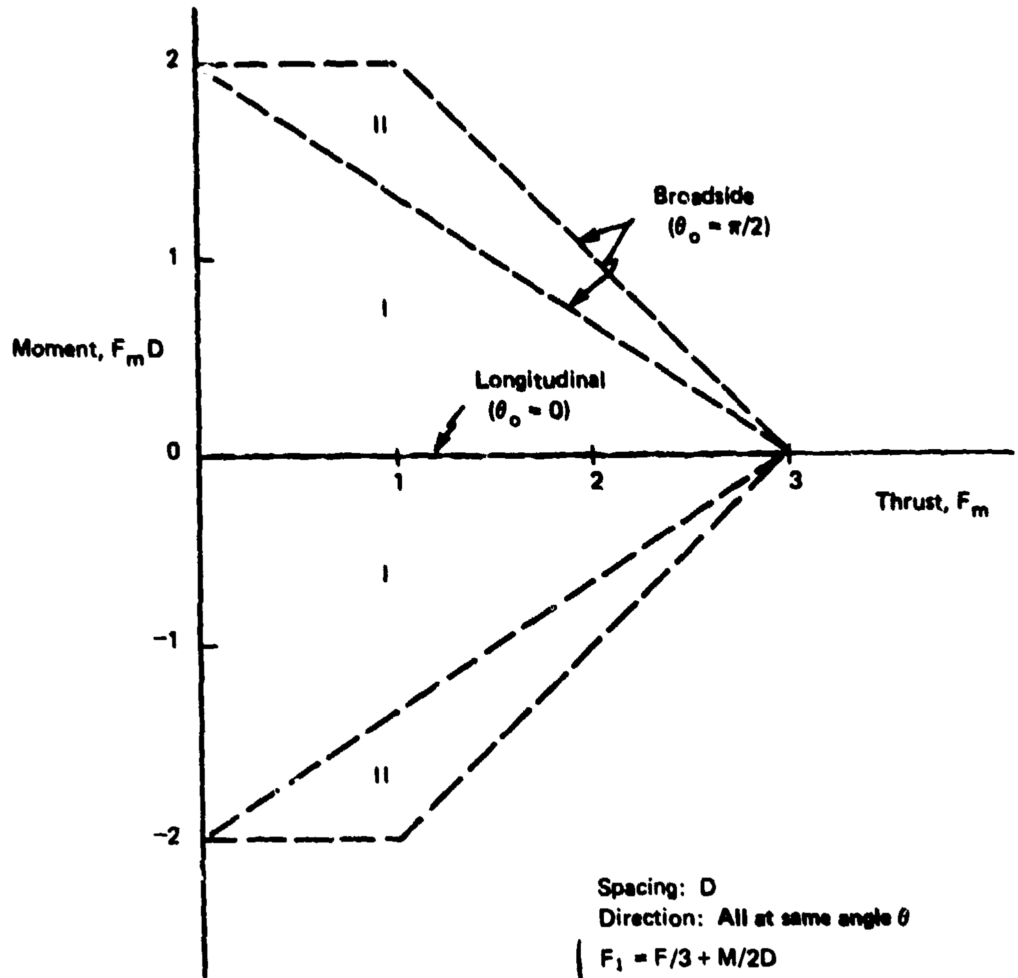
$$F_X = \sum^{N} F_i \cos\theta_i$$

$$F_Y = \sum^{N} F_i \sin\theta_i$$

$$\left.\begin{aligned} M &= \sum^{N} D_i F_i \sin\theta_i \\ F &= (F_X{}^2 + F_Y{}^2)^{1/2} \\ \theta &= \tan^{-1} F_Y/F_X \end{aligned}\right\} \text{3 Equations with 2 N Variables}$$

Figure 5-2

Basic Equations



Spacing: D
Direction: All at same angle θ

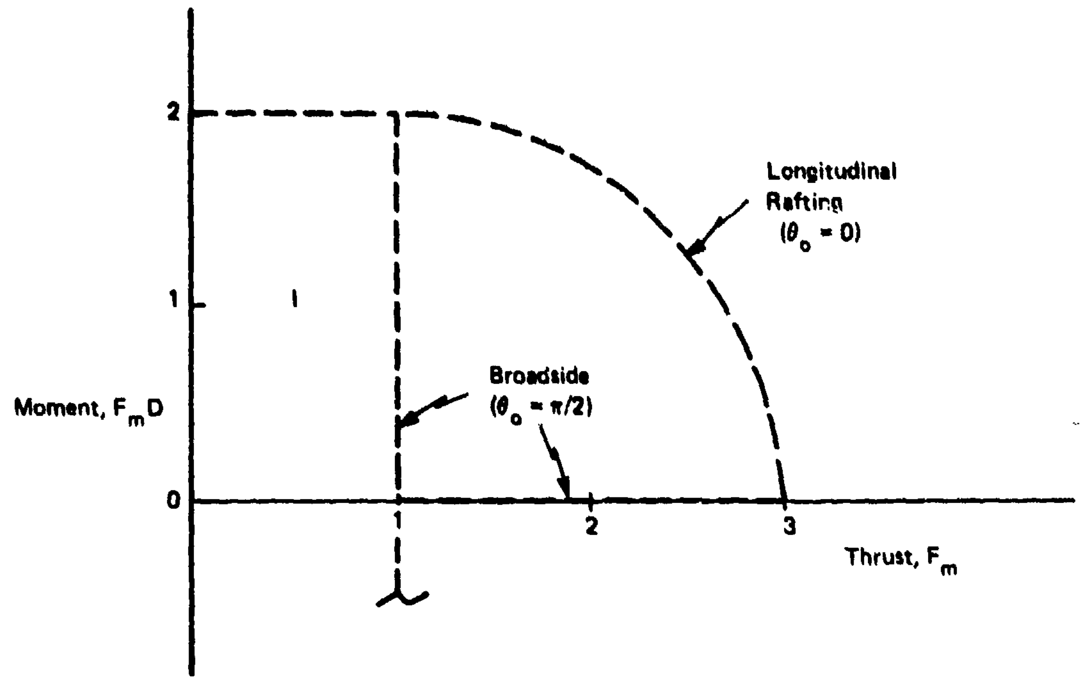
Within domain I $\begin{cases} F_1 = F/3 + M/2D \\ F_2 = F/3 \\ F_3 = F/3 - M/2D \end{cases}$

F_i smaller than maximum F_m.

If F_2 is not limited by $F_2 = F/3$, domain II is possible for use.

Figure 5-3

Envelope of Maximum Moment and Thrust -- Three Units at Same Angle



Spacing: D
Each Unit Power: F
Max. Power: F_m
Within domain I:

$$\theta_o = \tan^{-1} (\sin\theta_1 + \sin\theta_2 + \sin\theta_3)/(\cos\theta_1 + \cos\theta_2 + \cos\theta_3)$$

$$F_o = F\,[(\sin\theta_1 + \sin\theta_2 + \sin\theta_3)^2 + (\cos\theta_1 + \cos\theta_2 + \cos\theta_3)^2]^{1/2}$$

$$M_o = F\,D\,(\sin\theta_1 - \sin\theta_3)$$

Figure 5-4

Envelope of Maximum Moment and Thrust -- Three Units at Same Power

For four or more propulsion units, the forms of the equations would be similar to those indicated in these two examples, and the envelopes would show similar characteristics. However, there would be many more subalternatives within each domain, each of which would have to be evaluated to find the most effective or efficient combination. More sophisticated rules than those shown here may be applicable. As a result, the complexity of the controller would increase. Simplicity is, of course, highly desirable from first-cost, maintenance, and operational viewpoints.

5.4 MANUAL CONTROL

In the basic mode of control of a powered unit, the individual operator maintains direct and separate control over each of the two propulsion units. This mode is used when the craft is launched, when it is being connected to non-propelled bays, when it is employed as a boat, or when it is used to shift individual bays or disassemble a bridge. In all these cases the operator is in command of the movements of the craft and takes his general directions from a superior.

With the diesel and pumpjet system, the powered units are steerable through 360°, and shaft speeds match so that no clutch is required. Diesel engines have quite high minimum speeds; to place the craft in "neutral" without thrust or turning moment, the operator must balance the "idling" thrust from the pumpjets. When two opposing jets are to be balanced to zero, the directional settings will be reciprocals. For thrust in any one direction without turning moment, however, the settings will differ because the Schottel jets may not be on the centerline of the craft and the jets do not emerge from the center of the circular housing. The same equations shown in Figure 5-2 govern this situation, and the variations in settings will depend on the geometry of the final design.

The proper settings of the two propulsors can be learned by practice; alternatively, the settings can be determined by a microprocessor that calculates the two power settings and two direction settings when a demand for thrust and moment is set at a single control. If the settings are determined individually by the operator, he requires two sets of controls; if they are derived from a combined demand signal, he requires only a single set of controls.

In either case, since the propulsors can be steered in any direction, continuous smooth control through 360° is desirable. On manual control types using a shaft whose axial position determines power and whose rotational position determines direction, it is impossible to achieve such continuous angular motion without switching hand position. On a "joy stick" type of control, where the angle of inclination of a lever determines power and the direction of the plane of inclination determines direction, uninterrupted changes in the propulsors can be achieved by one hand. Therefore, for maneuvering purposes with separate manual controls, the latter seems more suited. For long-term operation as in a bridge where settings are adjusted infrequently, the axial/rotational shaft approach would suffice. For the case of the combined single control in which the individual propulsors are automatically adjusted, either type of control is suitable.

When the craft is used in a raft or bridge in coordination with similar units, command communications are somewhat different than when Bridge Erection Boats are used. In the latter case, the boats are behind or alongside the bays, and hand signals from the bridge/ferry commander are readily visible; with units equipped with integral propulsion, however, the operators are stationed at the ends of the craft, and visual communication may not always be practical. Amplified voice commands also may be unreliable because of engine and traffic noise.

If a multi-unit remote control system is adopted with three or four powered units interconnected by hard wire, the need for communications would be greatly reduced in the ferrying mode, since only one or two stations require direction. However, this represents only one potential mode and requirement.

The problems of stringing and assuring continuity of numerous telephone wires makes a hard-wire command network undesirable. Although the use of an additional UHF frequency in an already burdened RF spectrum is also undesirable, the best solution appears to lie in the use of portable transceivers at the boat control stations and at the commander's station. With hand-held equipment, the commander is free to use the best observation point, and the individual controllers can concentrate on their tasks without having to watch for visual signals. For single boat operators who are controlling the propulsors individually and manually, the additional task of responding on a transceiver may be difficult; equipment should be used that requires a minimum amount of manipulation.

In practice it may be found that a visual hand-signal system of command will continue to serve the purpose, but a UHF radio link would facilitate command in many situations and would cover all possible situations. While simplicity favors the visual system, the UHF system is a necessary adjunct for use when the former is inadequate; both should be available for use.

The question of the difficulty of manually controlling two pumpjets in a craft to achieve a desired net thrust and moment cannot be resolved here. Manual control has been used abroad, so it is possible. The difficulty of the task itself and the required personnel capabilities and learning ability both enter into the determination of training requirements. The question thus involves personnel policies, which are outside the scope of this study. Personnel numbers and qualifications for craft operators can obviously be reduced by the automation of control functions, but this advantage is offset by the higher initial cost of automated systems as well as increased maintenance costs, which include supplementary maintenance and repair personnel.

When the powered craft are separately controlled, whether by an individual propulsor control system or an automatic single control system, one operator per craft is required. This is a continuous function; one other part- or full-time person will be required for engine surveillance and adjustment, fueling, deckhand duties, and other intermittent actions. Relief personnel also must be considered.

When the powered craft are centrally controlled in groups, one operator is required for the central control position. A safety man on each craft is desirable, and this individual can perform the auxiliary functions noted above.

For a total of N powered units, which can be controlled individually or in groups of three, the following manpower comparison can be made. We assume that each powered unit can drive two non-powered bays and that three powered units would be necessary in a six-bay raft.

	Individual Control	Group Control
Operators	N	N/3
Assistants	N/2 to N	N/3 to N
Maint. Mechanic	N/3	N/3
Total	1.83 to 2.33 N	1.0 to 1.67 N

A manpower saving of at least 2/3 bodies per powered unit would be achieved by the use of automated group control.

Phase II

Organizational and Life-Cycle Costs of

Alternative Wet Bridge Systems

6. ORGANIZATIONAL AND LIFE-CYCLE COSTS OF ALTERNATIVE, VIABLE WET BRIDGE SYSTEMS

6.1 INTRODUCTION

This portion of the report concerns Phase II, which addresses the investigation of the organizational and life-cycle cost aspects of three alternative wet bridge system concepts, namely,

- The Current Ribbon Bridge,
- An Improved Ribbon Bridge utilizing integral propulsion half-bays to replace the present Construction Boats in the functional aspects of bridging as well as rafting, and
- An Improved Wet Bridge based upon three-ponton bays, each having its own integral propulsion.

The organizational aspects of the study were coordinated with the Technical Point of Contact at the Marine and Bridge Laboratory. The life-cycle cost aspects of Phase II were coordinated with the Cost Analysis Division of MERADCOM. Information on the current Ribgon Bridge system was obtained from the Readiness Project Officer, Troop Support and Aviation Command, St. Louis, Missouri.

Life-cycle costing was limited to a consideration of three principal elements of cost, expressed in fiscal year 1982 constant dollars for a ten-year period:

- Investment, namely, bridging systems and subsystem procurement;
- Comparative crew costs based upon military pay and allowances only, disregarding a loaded rate; and

- The operating and support costs of a bridging mission, calculated on the assumption that such missions occur (a) once per month and (b) eight times per month.

Other cost elements such as research and development were not included, either because of a lack of availability or the requirement for extensive engineering and detailed design, which were outside the scope of the task.

It was assumed that each alternative bridge would have a service life of not less than ten years (combined storage and mission use). The typical 24-hour mission for this bridge, which is capable of spanning a 120-m wet gap, was set forth as follows:

A 13-hour duration mission consisting of:

Approach march	0.5 hours or 10 miles
Construction (site preparation, launching and assembly of 400 feet of bridge)	1.0
Bridge operation (average traffic volume of 60 vehicles per hour to include at least five Class 60 vehicles)	9.0
Disassemble	2.0
Relocate	0.5
Total	13.0 hours

6.2 ACQUISITION COSTS

The acquisition costs are presented with their source for subsystems of the two new bridge concepts. Where necessary, estimates were made by Arthur D. Little, Inc., based upon conservative engineering and manufacturing practices. For some newer developments, such as the jet pumps, pricing information has been requested, but 1982 prices have been confirmed only for the SPJ-32 Schottel Pump-Jet (which will be $10,000 in FY1982 dollars).

6.2.1 Ribbon Bridge System

The acquisition costs for the Ribbon Bridge System in FY1982 dollars are presented in Table 6-1. These costs are based on using the new transporter expected to be available in 1983 and the new Bridge Erection Boat.

6.2.2 Improved Ribbon Bridge System with Integral Propulsion Half-Bays

The Improved Ribbon Bridge System utilizes the same subsystems as the Ribbon Bridge but replaces Bridge Erection Boats with integral propulsion half-bays for both bridge and rafting applications. The derivation of our cost estimate for the integral propulsion half-bay is presented below and in Table 6-2. The expected price for the half-bay in quantities of 100 or more and lot sizes of 50 is expected to be $78,259.

Multiple production costs (and prices) are based upon the following assumptions:

- Two factors are taken into account: (1) an exponential "learning curve" and (2) reduced amortization for special plant equipment for this production. It is assumed that production is in five-unit lots, and that the reductions are based upon lots rather than on individual units.
- A cost division is assumed, with exponents, as follows:

  Labor 30% and $.995^n$

  Material 50% and $.990^n$

  Overhead 20% and $.997^n$
- Plant savings are assumed to be .04 times the number of boats times dollars invested. The latter is taken at 5% of the contract cost.

Table 6-1

Acquisition Costs of Ribbon Bridge System

Subsystem or Module	Cost (FY1982 $)	Source*
Interior Bay	25,500	(1)
Ramp Bay	35,855	(1)
Transporter M 945 Chassis	67,725	(2)
Transporter Kit	12,982	(1)
Bridge Erection Boat (UK CSB)	150,000	(3)
Boat Cradle	7,200	(3)

*(1) Readiness Project Officer (Troop Support & Aviation Command)

(2) USATACOM

(3) Cost Analysis Division, MERADCOM

Table 6-2

Estimated Cost and Expected Price of Integral Propulsion Half-Bay for Improved Ribbon Bridge System

Structure:	4ST @ $4,500/ST	$ 18,000
Hinge:	Soss type; 11 ft @ $150/ft	1,650
Hydraulics:	Entire system including power takeoff, pump, piping accumulator, cylinders	4,000
Controls:	Two stations; cabling; controllers	5,000
Misc. Outfit:	Ventilation, navigation, bilge pumps, etc.	3,000
Propulsion:	2 x 150-bhp Deutz diesels @ $75/bhp, w/aux.	22,500
	2 x SPJ Schottel units	24,000
Total Unit Cost:		$ 78,150
	Tooling and Manufacturing Design and Engineering @ 5%	3,908
		$ 82,058
	Profit and Taxes @ 10%	8,206
Expected Price:		$ 90,264

Source: Arthur D. Little, Inc.

The combined reduction is approximated closely by the expression $C = cn\,[1.0 - .007\,(n-1)]$ where n is the number of five-boat units and c is the base cost for each boat. The results are as follows:

No. Half-Bays	% Savings	Cost	Expected Price
1	0	$ 82,058	$ 90,264
5	0	410,290	451,325
10	1.4	809,092	890,013
25	2.8	1,994,009	2,193,440
50	6.3	3,844,417	4,228,915
100	13.3	7,114,429	7,825,967

6.2.3 Improved Wet Bridge with Integral Propulsion Three-Ponton Bays

The acquisition costs for the subsystems of the Improved Wet Bridge are presented in Table 6-3. More detailed manufacturing costs on the two principal modules to be developed, namely, the interior bay and the ramp bay, are presented in Tables 6-4 and 6-5, respectively.

6.3 CREW COSTS

The annual crew costs were obtained from the Cost Analysis Division, MERADCOM, and represent military pay and allowances only. They do not represent a loaded rate. These man-year costs are presented in Table 6-6.

The crew for the Current Ribbon Bridge numbers 60. The crew for the Improved Ribbon Bridge with integral propulsion half-bays is only 51 because of the reduction of transporters from 27 to 24 and the reduction of Bridge Erection Boats from 8 to 2--one a safety boat, the other a marshalling boat. Crew requirements are further reduced to 48 for the Improved Wet Bridge with integral propulsion three-ponton bays, since only 19 transporters are needed.

Table 6-3

Acquisition Costs for Improved Wet Bridge System with Integral Propulsion Three-Ponton Bays

Subsystem or Module	Cost (FY1982 $)	Source
Interior Bay with Integral Propulsion	98,759	Table 6-4
Ramp Bay with Integral Propulsion	89,869	Table 6-5
Transporter, 10-Ton HEMTT Chassis	108,600	Budgetary Price, Oshkosh Truck Corporation, Oshkosh Wisconsin 54903
Transporter Kit	14,950	ADL (15% increase over Ribbon Bridge TK Cost)

Table 6-4

Estimated Cost of Three-Ponton Interior Bay

Subsystem Module or Item	Description or Quantity	Unit Cost ($)	Cost (FY1982 $)
Structure	6.91 ST	4,250/ST	29,368
Hinge: Soss Type	20 ft	150/ft	3,000
Hydraulics	Entire system		4,000
Controls	Two stations, cabling and controllers		5,000
Auxiliary Equipment	Fuel tanks, combustion and ventilation sub-system, bilge pump, covers, hoists		6,000
Propulsion Subsystem: Deutz Diesel F6L 912	2	5,569	11,138
Schottel Pump-Jet SPJ 32 or SPJ 20	2	10,000	20,000
Installation	2	3,500	7,000

Unit Quantities, Lots of 50
Ultimate 2,000

Cost	85,506
Tooling and Manufacturing Design and Engineering @ 5%	4,275
	89,781
Profit and Taxes @ 10%	8,978
Estimated Price	98,759

Table 6-5

Estimated Cost of Three-Ponton Ramp Bay

Subsystem Module or Item	Description or Quantity	Unit Cost ($)	Cost (FY1982 $)
Structure	6.73 ST	6,130/ST	41,240
Hinges: Soss Type	20 ft	150/ft	3,000
Hydraulics	Entire System Ponton positioning and securing, ramp elevating		6,000
Controls	Two stations, cabling and controllers		4,000
Auxiliary Equipment	Fuel tanks, combustion and ventilation subsystem, bilge pump, covers, hoists		4,000
Propulsion Subsystem: Deutz Diesel F6L 912	1		5,569
Schottel Pump-Jet SPJ 32	1		10,000
Installation	1		4,000
Unit Quantities, Lots of 10 Ultimate 200		Cost	77,809
Tooling and Manufacturing Design and Engineering @ 5%			3,890
			81,699
		Profit and Taxes @ 10%	8,170
		Estimated Price	89,869

Table 6-6

Annual Crew Costs

(Military Pay & Allowances Only)

Rank	Cost (FY1982 $)
1st Lt.	24,902
E7	22,423
E6	18,788
E5	15,655
E4	13,316
E3	11,705

6.4 OPERATING AND SUPPORT COSTS

Operating and support costs include operational and maintenance costs (excluding the bridging crews). Personnel costs are included in maintenance operations involving overhaul. These costs were estimated for two conditions: one mission per month and eight missions per month. The latter represents a high degree of activity, such as for initial training with new bridging and rafting systems.

6.4.1 Current Ribbon Bridge

A review was made of the Logistic Management Analysis Quarterly Summary (Parts A and B) for 1 November 80-31 January 81 on the Ribbon Bridge System; this included:

- Truck (5-ton)
- Transporter
- Bridge erection boat
- Boat cradle
- Interior bay
- Ramp bay

The analysis was made by the US Army Troop Support and Aviation Materiel Readiness Command, Directorate for Product Assurance, Systems Performance Assessment Division. The basic findings listed below applied to low usage at both Fort Hood and Fort Lewis.

Combined Scheduled and Unscheduled Maintenance

Item	Maintenance Man-Hours per Operating Hour
5-ton truck	0.13
Bridge erection boat	0.45

No maintenance requirements were reported for the non-powered modules (interior bay, ramp bay, transporter kit, and boat cradle).

Operating and maintenance costs for the more static modules were estimated at a spares usage of 2% per year, regardless of the frequency of missions.

The annual operating and support costs for the Current Ribbon Bridge System are presented in Table 6-7.

6.4.2 Improved Ribbon Bridge with Integral Propulsion Half-Bay

The operating and support cost for the integral propulsion half-bay was estimated to be approximately the same as that for the Bridge Erection Boat, or $26.02/OH. The annual cost for spares, however, would be less; it was estimated at 2% of the cost of the static components, or $700 annually. Hence, the annual costs were estimated as follows:

- Annual cost for one mission/month $ 4,447
- Annual cost for eight missions/month $40,676

6.4.3 Improved Wet Bridge with Integral Propulsion Three-Ponton Bays

The annual operating and support cost for the subsystems and modules of the Improved Wet Bridge are presented in Table 6-8.

6.5 COMPARATIVE LIFE-CYCLE COSTS OF CURRENT RIBBON BRIDGE, IMPROVED RIBBON BRIDGE, AND IMPROVED WET BRIDGE SYSTEMS

This section discusses separately the three cost elements that comprise life-cycle cost, namely,

- Acquisition costs,
- Crew costs, and
- Operating and support mission costs.

The final subsection, 6.5.4, summarizes the ten-year life-cycle costs. Both the Improved Ribbon Bridge with integral propulsion half-bays and the

Table 6-7

Annual O&S Costs for Current Ribbon Bridge System

Subsystem or Module	Annual O&S Cost (FY 1982 $)		Source*	Assumptions and Comments*
	1 Mission/Month	8 Missions/Month		
Interior Bay	510	510	(1)	(3)
Ramp Bay	718	718	(1)	(3)
Transporter - 5 ton 945 chassis	520.80	4,166.40	(2)	Overhaul cost/OH estimated to be $1.03. Operating cost/hr, including diesel fuel at $1.29/gal is $9.82, for a total cost of $10.85/truck hr.
Transporter Kit	390	390	(1)	(3)
Bridge Erection Boat (UK CSB)	6,747	32,975	(2)	Estimate based upon heavy-duty centrifugal pump, diesel powered. Overhaul cost for each diesel engine/Pump-Jet combination estimated at $6.92/OH; operating cost estimated at an additional $6.09/hr, for a total of $26.02/boat hr. Duration of operation per boat estimated at 12 hr/mission. Hence, mission cost/boat = $312.24.
Boat Cradle	144	144	(1)	(3)

*(1) MERADCOM and Arthur D. Little, Inc.
(2) "Cost Reference Guide for Construction Equipment," Nielsen/Dataquest, Inc.
(3) Annual usage of spares equal to 2% of procurement cost.

Table 6-8

Annual O&S Costs for Improved Wet Bridge with Integral Propulsion Three-Ponton Bays

Subsystem or Module	Annual O&S Cost (FY 1982 $) 1 Mission/Month	8 Missions/Month	Source, Assumptions and Comments
Interior Bay	3,799.56	23,236.48	The O&S costs were based upon those for a heavy-duty centrifugal pump powered by an 88-hp diesel engine.* The overhaul cost was estimated to be $6.92/OH, and the operating cost at 2 gal/hr/engine was estimated to be $4.11/hr. Hence, the total cost for the interior bay was estimated to be $22.06/OH, not including spare parts for the static subsystems. These were estimated at 2% of the procurement cost of $50,000 for these static and structural subsystems, or $1,000 per annum.
Ramp Bay	2,752.32	13,618.56	Based upon use of a single diesel engine/Pump-Jet and a spare parts cost of 2% of the procurement cost of the static subsystem, or $1,200.
Transporter - 10-ton HEMTT Chassis	710.40	5,683.20	The transporter was considered to operate 4 hr per mission. Overhaul cost estimated at $1.41/OH and operating cost estimated at $13.39/OH, for a total truck-hour cost of $14.80.*
Transporter Kit	450	450	Annual spares usage estimated to be 3% of the procurement cost, or $450.**

*Source: "Cost Reference Guide for Construction Equipment," Nielsen/Dataquest, Inc.
**Source: MERADCOM and Arthur D. Little, Inc.

Improved Wet Bridge with integral propulsion three-ponton bays proved more cost-effective than the Current Ribbon Bridge System. The Improved Ribbon Bridge is cost-effective for all three cost elements. The Improved Wet Bridge is, as expected, higher in acquistion costs and mission O&S costs and lowest in crew costs, but its improved bridge/rafting performance and land mobility over rough terrain more than compensates for the marginally increased acquisition and mission O&S costs. Its total life-cycle cost for typical mission activities is less than that of the Current Ribbon Bridge System.

6.5.1 Acquisition Costs

The acquisition costs for each of the three bridge systems are presented in Table 6-9. By eliminating the need for six Bridge Erection Boats and replacing them with integral propulsion half-bays, and eliminating the need for three transporters and transporter kits, it was possible to reduce the acquisition cost for the Improved Ribbon Bridge by 20% in comparison with the Current Ribbon Bridge System. For the Improved Wet Bridge, the additional cost of integral propulsion in the bays and the higher cost of the 10-ton high-mobility transporter increased the acquisition cost of this system by 18% over the Current Ribbon Bridge. However, the effectiveness of the Improved Wet Bridge System far outweighs this additional cost.

6.5.2 Crew Costs

The annual crew costs (including military pay and allowances only) are presented for each of the three bridging systems in Table 6-10. The crew cost is least for the Improved Wet Bridge because of the elimination of six transport drivers and six erection boat operators. For the ten-year life-cycle crew cost, the reduction for the Improved Ribbon Bridge with

Table 6-9

Comparative Acquisition Cost

- 120 Meters of Wet Bridge
- FY '82 Constant Dollars

Bridge Subsystem or Module	Current Ribbon Bridge System: Quantity Required	Current Ribbon Bridge System: Unit Cost ($000)	Current Ribbon Bridge System: Total Cost ($000)	Improved Ribbon Bridge with Inte[g] Propulsion Half-Bays: Subsystem Changes	Improved Ribbon Bridge with Inte[g] Propulsion Half-Bays: Quantity Required	Improved Ribbon Bridge with Inte[g] Propulsion Half-Bays: Unit Cost ($000)
Interior Bay	17	25.5	433.5		14	25.5
Ramp Bay	2	35.9	71.8		2	35.9
Half Bay w/IP	-	-	-	Half Bay w/IP	6	78.3
Transporter						
5-ton M945 chassis	27	67.7	1,827.9		24	67.7
10-ton HEMTT chassis						
Transporter Kit	27	13.0	351.0		24	13.0
Bridge Erection Boat (UK CSB)	8	150.0	1,200.0		2	150.0
Boat Cradle	8	7.2	57.6		2	7.2
Subtotal			3,941.8			
Spares at 5% of Procurement Cost			197.1			
Total			4,138.9			

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sition Costs

Wet Bridge
nt Dollars

with Integral f-Bays		Improved Wet Bridge with Integral Propulsion Three-Ponton Bays			
Unit Cost ($000)	Total Cost ($000)	Subsystem Changes	Quantity Required	Unit Cost ($000)	Total Cost ($000)
25.5	357.0	Interior Bay w/IP	17	98.9	1,679.6
35.9	71.8	Ramp Bay w/IP	2	89.9	179.8
78.3	469.8		-	-	-
67.7	1,624.8		2	67.7	135.4
			19	108.6	2,063.4
13.0	312.0		19	15.0	285.0
150.0	300.0		2	150.0	300.0
7.2	14.4		2	7.2	14.4
	3,149.8				4,657.6
	157.5				232.9
	3,307.3				4,890.5

2

Table 6-10

Comparative Crew Costs

- 120 Meters of Wet Bridg
- FY '82 Constant Dollars
- Military Pay and Allowa

Rank	Current Ribbon Bridge System			Improved Ribbon Bridge with Propulsion Half-Bay		
	Crew Required	Man-Year Cost ($000)	Total Annual Cost ($000)	Crew Required	Man-Year Cost ($000)	T
1st Lt.	2	24.9	49.9	2	24.9	
E7	2	22.4	44.8	2	22.4	
E6	3	18.8	56.4	3	18.8	
E5	7	15.7	109.9	7	15.7	
E4	29	13.3	385.7	24	13.3	
E3	17	11.7	198.9	13	11.7	
Total	60		845.5	51		
Ten Year Crew Cost			8,455.0			

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-10

Crew Costs

's of Wet Bridge
nstant Dollars
Pay and Allowances Only

on Bridge with Integral lsion Half-Bays		Improved Wet Bridge with Integral Propulsion Three-Ponton Bays		
Man-Year Cost ($000)	Total Annual Cost ($000)	Crew Required	Man Year Cost ($000)	Total Annual Cost ($000)
24.9	49.8	2	24.9	49.8
22.4	44.8	2	22.4	44.8
18.8	56.4	3	18.8	56.4
15.7	109.9	7	15.7	109.9
13.3	319.2	23	13.3	305.9
11.7	152.1	11	11.7	128.7
	732.2	48		695.5
	7,322.0			6,955.0

2

integral propulsion half-bays is 13%. The reduction in ten-year life-cycle crew costs for the Improved Wet Bridge over the Current Ribbon Bridge is 18%.

6.5.3 Mission Operating and Support Costs

MERADCOM has defined a typical 24-hour mission for the construction, operation and disassembly of a Wet Bridge over a 120-m water gap as consisting of 13 hours of activity. A breakdown by task was given in Section 6.1. The erection boats and the integral propulsion bays for the new bridges were assumed to operate for an average of 12 hours per mission, and a transporter was assumed to operate for an average of 4 hours. Table 6-11 compares the annual O&S costs of the three bridge systems for one mission per month, which would be typical for peacetime training.

The annual mission cost is least for the Improved Ribbon Bridge with integral propulsion half-bays, principally because the operating cost of the half-bay would be much less than that of a bridge erection boat. Another (relatively small) difference in cost results from the reduction of transporters and transporter kits from 27 to 24.

The highest operating cost is for the Improved Wet Bridge with Integral Propulsion Three-Ponton Bays. This reflects the O&S costs of the 17 interior bays and 2 ramp bays, all of which have integral propulsion. Transporter costs, even with the 10-ton HEMTT transporters, are less than those for the 5-ton M945 transporters used with the Current Ribbon Bridge because of the reduction from 27 transporters to 21.

Overall, the reduction in mission O&S costs was 21% for the Improved Ribbon Bridge in comparison with the Current Ribbon Bridge. The mission O&S costs of the Improved Wet Bridge, however, were 19% above those of the Current Ribbon Bridge.

Table 6-11

Comparative Annual Operating an

- 120 Meters of Wet Bridge
- FY '82 Constant Dollars
- Exclusive of Crew Costs
- Training Frequency: 1 Missi

	Current Ribbon Bridge System			Improved Ribbon Bridge wit Propulsion Half-Ba		
Bridge Subsystem or Module	Module Quantity Required	O&S Annual Cost/ ($)	Total Annual Cost ($000)	Subsystem Changes	Module Quantity Required	O
Interior Bay	17	510	8.7		14	
Ramp Bay	2	718	1.4		2	
Half-Bay w/IP	-	-	-	Half Bay w/IP	6	
Transporter						
5-ton M945 chassis	27	520.80	14.1		24	
10-ton HEMTT chassis	-	-	-		-	
Transporter Kits	27	390	10.5		24	
Bridge Erection Boats (UK CSB)	8	6,747	54.0		2	
Boat Cradles	8	144	1.2		2	
Total			89.9			
Ten Year Mission Cost			899.0			

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6-11

ting and Support (O&S) Costs

ge
ra
ts
1 Mission per Month

y d	ge with Integral alf-Bays O&S Annual Cost/ ($)	Total Annual Cost ($000)	Improved Wet Bridge with Integral Propulsion Three-Ponton Bays Subsystem Changes	Module Quantity Required	O&S Annual Cost/ ($)	Total Annual Cost ($000)
	510	7.1	w/IP	17	3,779.56	64.3
	718	1.4	w/IP	2	2,752.32	5.5
	4,447	26.7		-	-	-
	520.80	12.5		2	520.80	1.0
	-	-		19	710.40	13.5
	390	9.4		19	450	8.6
	6,747	13.5		2	6,747	13.5
	144	0.3		2	144	0.3
		70.9				106.7
		709.0				1,067.0

2

To take into account the intensive training needed for a new bridge system, MERADCOM requested that a frequency of eight missions per month also be considered. (This would not be normal for peacetime training.) The effect is shown in Table 6-12: annual O&S costs of the integral propulsion bays, the transporters, and the bridge erection boats are increased by factors ranging from about five to eight.

6.5.4 Summary of Ten-Year Life-Cycle Cost

The ten-year life-cycle costs representative of current mission operating procedures are summarized in Table 6-13. Acquisition costs are lowest for the Improved Ribbon Bridge System and highest for the Improved Wet Bridge System (although the higher performance of the latter system more than compensates for this cost). Crew costs are lower for both the Improved Ribbon Bridge System and the Improved Wet Bridge System. Mission O&S costs are lowest for the Improved Ribbon Bridge and somewhat higher for the Improved Wet Bridge System.

In terms of the ten-year totals, the Improved Ribbon Bridge offers a 16% saving over the Current Ribbon Bridge, and the Improved Wet Bridge offers a 4% saving.

A similar tabulation for a training frequency of eight missions per month is shown in Table 6-14. This frequency, as mentioned above, would be abnormal for peacetime, but the correspondingly higher O&S costs are more in keeping with the acquisition costs of the respective systems.

6.6 SUMMARY OF PERFORMANCE CHARACTERISTICS

Table 6-15 summarizes the performance characteristics of the three bridge systems covered in Phase I. These characteristics pertain to a 120-m wet bridge mode and three rafting modes (three-bay, four-bay, and five-bay).

Table 6-12

Comparative Annual Operating and Sup

- 120 Meters of Wet Bridge
- FY '82 Constant Dollars
- Exclusive of Crew Costs
- Training Frequency: 8 Missions p

	Current Ribbon Bridge System			Improved Ribbon Bridge with Int Propulsion Half-Bays		
Bridge Subsystem or Module	Module Quantity Required	O&S Annual Cost/ ($)	Total Annual Cost ($000)	Subsystem Changes	Module Quantity Required	O&S An Cost ($)
Interior Bay	17	510	8.7		14	51
Ramp Bay	2	718	1.4		2	71
Half-Bay w/IP	-	-	-	w/IP	6	30,67
Transporter						
5-ton M945 chassis	27	4,166.40	112.5		24	4,16
10-ton HEMTT chassis	-	-	-		-	-
Transporter Kits	27	390	10.5		24	39
Bridge Erection Boats (UK CSB)	8	32,975	263.8		2	32,97
Boat Cradles	8	144	1.2		2	14
Total			398.1			
Ten Year Mission Cost			3,981.0			

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e 6-12

ting and Support (O&S) Costs

idge
ars
ts
8 Missions per Month

idge with Integral Half-Bays			Improved Wet Bridge with Integral Propulsion Three-Ponton Bays			
le ity red	O&S Annual Cost/ ($)	Total Annual Cost ($000)	Subsystem Changes	Module Quantity Required	O&S Annual Cost/ ($)	Total Annual Cost ($000)
	510	7.1	w/IP	17	23,236.58	395.0
2	718	1.4	w/IP	2	13,618.56	27.2
6	30,676	184.1	w/IP	-	-	-
4	4,166.40	100.0		2	4,166.40	8.3
	-	-		19	5,683.20	108.0
24	390	9.4		19	450	8.6
2	32,975	66.0		2	32,975	66.0
2	144	0.3		2	144	0.3
		368.3				613.4
		3,683.0				6,134.0

2

Table 6-13

Ten Year Life Cycle Cost Summary for Low Mission Frequency

- 120 Meters of Wet Bridge
- FY '82 Constant Dollars
- Military Pay and Allowances Only
- Training Frequency: One (1) Mission per Month

Cost Element	Current Ribbon Bridge System ($000)	Improved Ribbon Bridge System with Integral Propulsion Half-Bays ($000)	Improved Wet Bridge with Integral Propulsion Three-Ponton Bays ($000)
Acquisition (including spares)	4,139	3,307	4,891
Crew	8,455	7,322	6,955
Mission O&S	899	709	1,067
Total	13,493	11,338	12,913

Table 6-14

Ten Year Life Cycle Cost Summary for High Mission Frequency

- 120 Meters of Wet Bridge
- FY '82 Constant Dollars
- Military Pay and Allowances Only
- Training Frequency: Eight (8) Missions per Month

Cost Element	Current Ribbon Bridge System ($000)	Improved Ribbon Bridge System with Integral Propulsion Half-Bays ($000)	Improved Wet Bridge with Integral Propulsion Three-Ponton Bays ($000)
Acquisition (including spares)	4,139	3,307	4,891
Crew	8,455	7,322	6,955
Mission O&S	3,981	3,683	6,134
Total	16,575	14,312	17,980

Table 6-15

Performance Trade-Off A

Performance Characteristics for Operational Modes	Current Ribbon Bridge System	Improved Ribbon Brid with Integral Propulsi
1. 120 Meter Wet Bridge Mode.		
a. Mission crew required.	60	51
b. Bridge erection boats required.	8	2
c. Transporters required.	27	24
d. Available thrust.	Forward -6 x 4500 = 27,000 Reverse -6 x 2200 = 13,200	Omnidirectional Forward - 12 x 2248 = 2 Reverse -12 x 2248 = 2
e. Available interior bay buoyancy with roadway awash.	~32.0 ST	~32.0 ST (also two integral pro
f. Launchability and recoverability.	Bay launched folded. Unfolded in water by its own buoyancy. Controlled and towed to bridge site by Bridge Erection Boat. For recovery, steps are reversed.	Half-bay unfolded on t own power and secured Half-bay launched from Half-bay connects to o interior full bays, t site and connects to more interior bays. For recovery, steps ar
g. Operability.	Requires up to six Bridge Erection Boats for live anchoring (crews of two per boat); anchoring crew of 12.	Requires up to six int bays for live anchori bay); anchoring crew
2. Rafting.		
a. Mission crew required. (Not including raft commander.)		
3 bay (ramp, interior, ramp)	4	
4 bay (ramp, 2 interior, ramp)	4	
5 bay (ramp, 3 interior, ramp)	4 for 2 boats, 6 for 3 boats	
b. Bridge erection boats required (3, 4, and 5 bay).	2 to 3	
c. Transporters required.		
3 bay	5	
4 bay	6	
5 bay	7 or 8	

le 6-15

Trade-Off Analysis

Ribbon Bridge System al Propulsion Half Bays	Improved Wet Bridge System with Integral Propulsion Three Ponton Bays
51	48
2	2
24	21
onal x 2248 = 26,976 x 2248 = 26,976	Omnidirectional Maximum 36 x 1574 = 56,664 @ 1800/module 19 x 1800 = 34.200 @ 1300/module 19 x 1300 = 24,700
~32.0 ST ntegral propulsion half bays)	~34.1
folded on transporter by its and secured. unched from transporter. nnects to one or more ramp or ull bays, tows bays to bridge onnects to bridge or returns for ior bays. y, steps are reversed.	Both ramp and interior bays are unfolded on transporter by their own power and secured. Bays launched from transporters. Bays maneuver to bridge site under own power. Depending upon controllability, ramp bay may require early connection to interior bay. Bays are interconnected to form the bridge. For recovery, steps are reversed.
to six integral propulsion half-ive anchoring (crew of one per half-oring crew of six.	Requires one operator per four integral propulsion bays; anchoring crew of five.
4	1
2	1
2 for 2 half bays 4 for 4 half bays	2
0	0
4	3
5	4
6 or 7	5

2

Performance Characteristics for Operational Mode	Current Ribbon Bridge System		Improved Ribbon Bridge Sy with Integral Propulsion Ha
d. Available thrust. (pounds)	Forward	Reverse	Omnidirectional
3 bay	9,000	4,400	4,496
4 bay	9,000	4,400	4,496
5 bay	9,000 to 13,500	4,400 to 6,600	4,496 to 8,992
e. Raft capacity (Military Load Class) based upon:			
0.1 m stillwater roadway freeboard			
3 bay	20		20
4 bay	40+		40+
5 bay	60+		60+
0.4 m stillwater bow freeboard			
3 bay	20		20
4 bay	40+		40+
5 bay	60+		60+

e 6-15 (Cont'd.)

on Bridge System ropulsion Half-Bays	Improved Wet Bridge System with Integral Propulsion Three-Ponton Bays		
ctional	Omnidirectional		
	1300 Lbs/Module	1800 Lbs./Module	Max. 1574 Lbs./Pump-Jet
496	3,900	4,948	6,296
496	5,200	6,748	9,444
496 to 992	6,500	8,548	12,592
20		30+	
40+		60+	
60+		90+	
20		10+	
40+		30+	
60+		50+	

2

In practically all performance categories the Improved Ribbon Bridge System with integral propulsion half-bays is superior to the Current Ribbon Bridge System. In a few categories (e.g., capacity as a bridge or as a raft) the two systems are equal. The only inferior operational characteristic of the Improved Ribbon Bridge is in rafting, since the forward thrust of a half-bay is only half that of the new UK Bridge Erection Boat.

The Improved Wet Bridge System with integral propulsion three-ponton bays is superior in every characteristic except the forward available thrust of a three-bay raft. It is superior in its omnidirectional thrust, however, and has approximately 50% more thrust in the reverse direction. The only other inferior attribute of the Improved Wet Bridge is its bow freeboard in relation to the roadway freeboard. The raft and bridge capacity with respect to roadway freeboard is far superior to that of the Current Ribbon Bridge.

APPENDIX A

WEIGHT AND BUOYANCY CALCULATIONS

SUMMARY

A review of the Current Ribbon Bridge drawings (specifically, -4003, Ponton, Interior, Roadway) indicates that of the 12,000-lb total weight of a complete interior bay, approximately 6700 lb is devoted to moment and tire-load reaction structure. The remaining weight consists of the bows, connections, local stiffeners, and other parts.

If the present bridge can carry Class 60 (max. wheeled vehicle load of 70 ST) and the Improved Wet Bridge must carry Class 70 (80.5 ST), a prudent estimate of the required increase in "structural" weight would be the same as the increase in actual load, namely, 15%. The increased depth of the bridge section (from 29 to approximately 38 in.) would apparently increase the structural "section modulus" <u>without</u> an increase in weight; however, in deep sections web buckling can control material thickness. Without detailed knowledge of point loads, point of application of worst-case loads, or worst-case load distribution, it is impossible to refine the estimate of weight increase.

In the Improved Wet Bridge, the wheel or track loads will (for the worst case) cross the bow-to-interior longitudinal joint, thus partially loading the bow. This longitudinal hinge joint must be capable of sustaining the associated shear transfer loads. Each of the two hinges and two lower connection mechanisms is estimated to be equivalent to a 2-1/2 inch square aluminum bar extending for a bay length of 275.6 in. (7.0 m). This total weight is 688 lb.

Thus, the total increase in weight will be 6700 x 15% plus 688, or approximately 1700 lb. Since this figure reflects only structural differences, it should be considered the minimum weight increase.

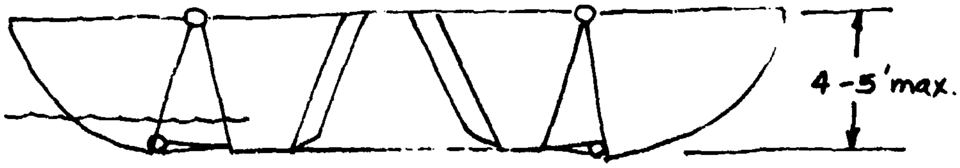
- From TM 5-5420-209-12:

  5-ton Transporter can carry 6-ton load

  Ribbon Bridge has Class 60 capability in currents up to 8 ft/s

- From Report 2298:

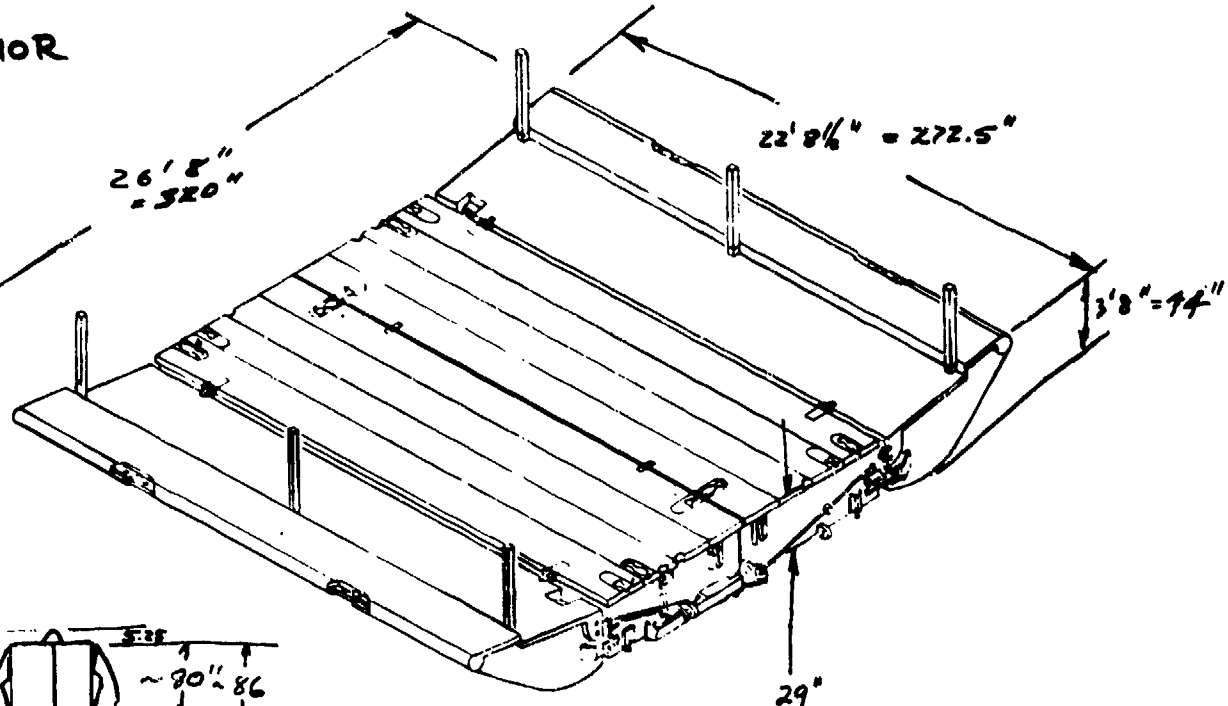
  Wet Bridge launch from WVL

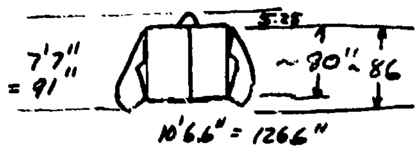


- From Trilateral Design Report, Appendix C:

	Weight of Tank	Max. Load	Single Axle	Critical Tire Load and Tire Size	Tank Width
Class 60	60	70	23	20,000 lb on 24.00 K 29	132"
Class 70	70	80.5	25.5	Ditto	138"
	↓	↓	↓		↓
	+16.7%	+15%	+11%	No change	+4.5%

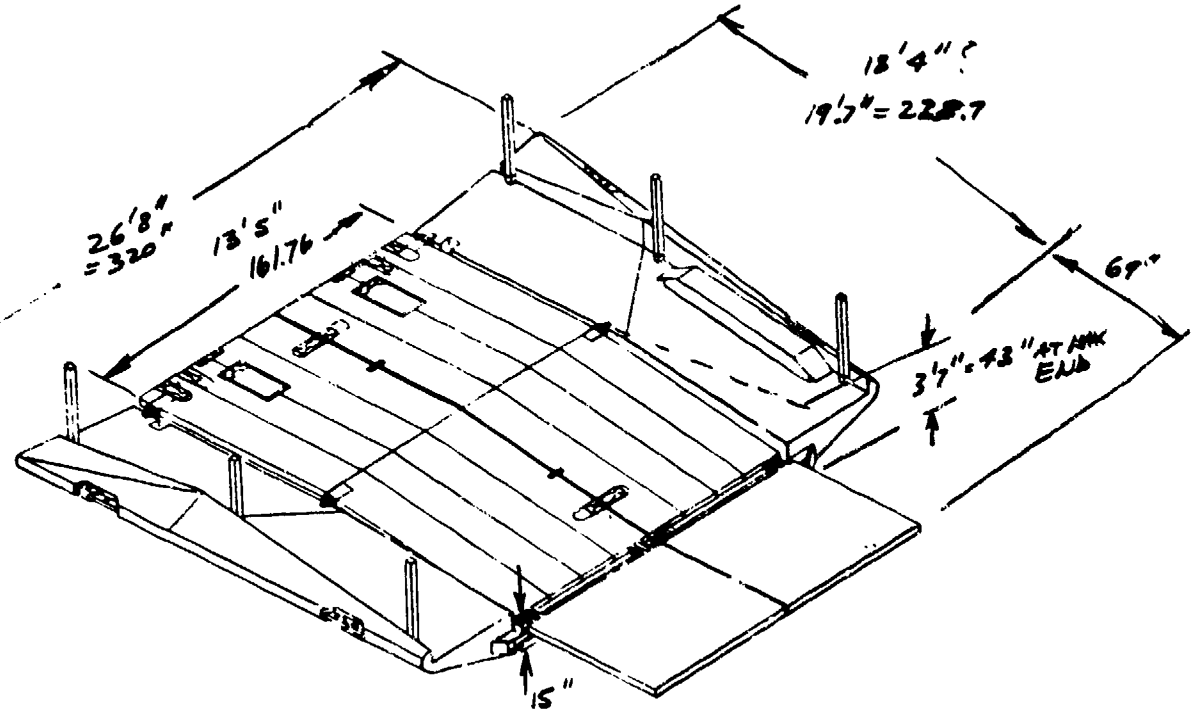
INTERIOR BAY

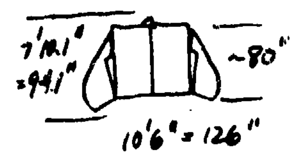




12,000 LBS 1817 FT³ 6.604 #/FT³
OVERALL, FOLDED

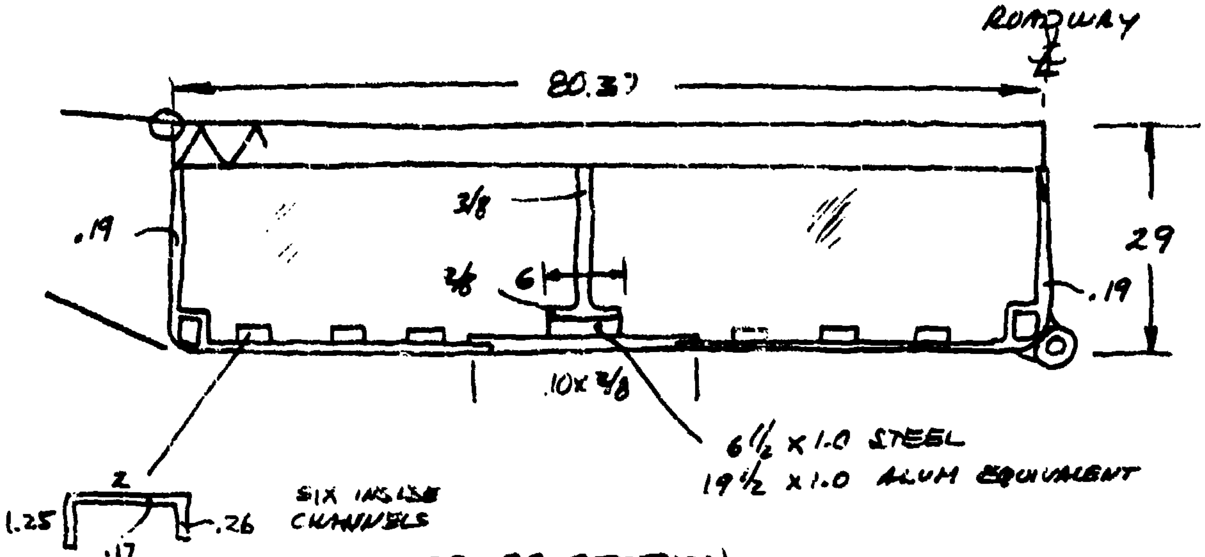
RAMP BAY





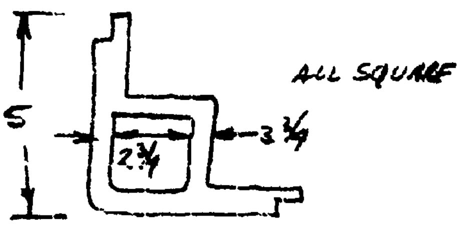
11,700 LBS 1566 FT³ 7.471 #/FT³
OVERALL, FOLDED

CURRENT RIBBON BRIDGE

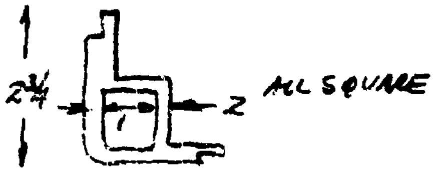


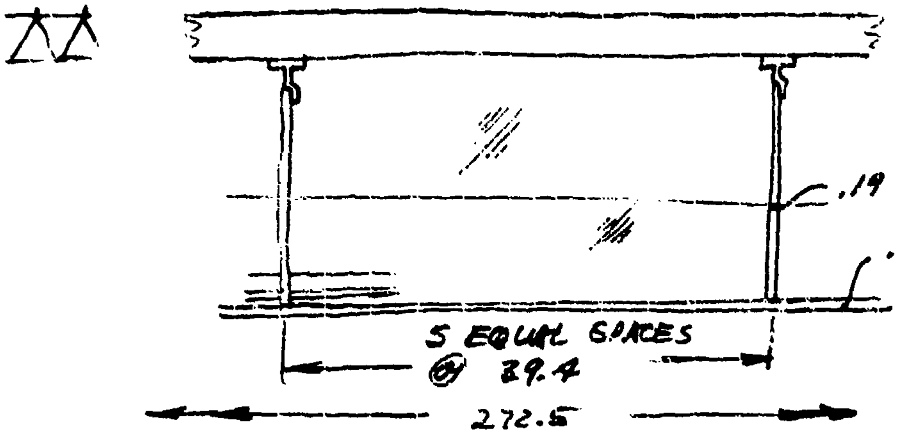
CROSS SECTION

CORNERS ON ENDS



CORNERS ON ROADWAY
ESTIMATED

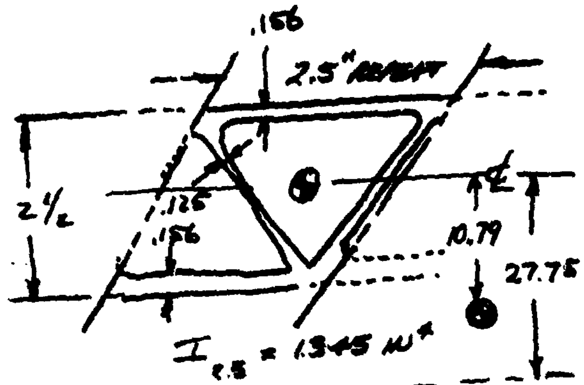




LONGITUDINAL SECTION

CG_{TOTAL} shown by ●

DECKING (ONE)



CROSSSECTIONAL AREA OF EACH REPEAT:

$$2(2.5)(.156) + 2(2.5/\sin 60)(.125)$$

$$= 0.78 + .722$$

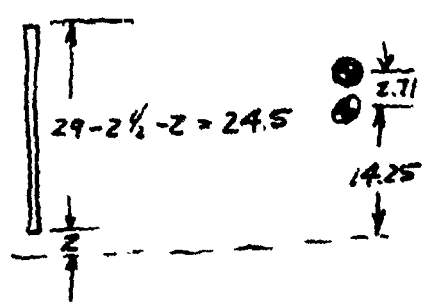
$$= 1.50\ IN^2 \text{ PER 2.5" OF ROAD WAY WIDTH}$$

TOTAL DECK AREA $= 1.5\left(\frac{80.37}{2.5}\right) = 48.2\ IN^2$ WITH C.G.

LOCATED AT $29 - \frac{2.5}{2} = 27.75$" ABOVE REF. AXIS

$$I_{CG} = \left[\frac{(2.5)^4}{12} - \frac{(2.5-2[.156])^4}{12}\right]\frac{80.37}{2.5} = 43.25\ IN^4$$

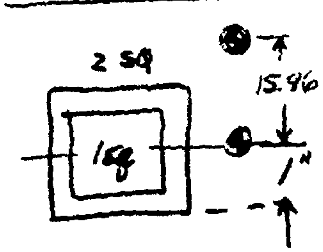
SIDE PLATES (TWO)



$$AREA = 24.5(.190) = 4.66\ IN^2$$

$$I_{CG} = \frac{(.190)(24.5)^3}{12} = 232.8\ IN^4$$

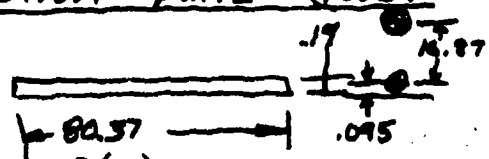
CORNERS (TWO)



$$AREA = 2^2 - 1^2 = 3\ IN^2$$

$$I_{CG} = \frac{2^4}{12} - \frac{1^4}{12} = 1.25\ IN^4$$

BOTTOM PLATE (TWO)

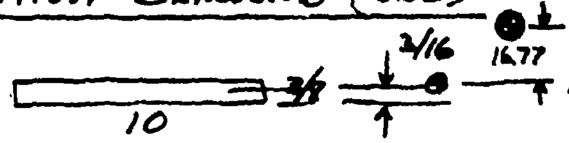


−2(2)
−10
(66.37)÷2 = 33.2

$$AREA = \frac{66.37}{2}(.19) = 6.3\ IN^2$$

$$I_{CG} = \frac{(33.2)(.19)^3}{12} = 0.019\ IN^4$$

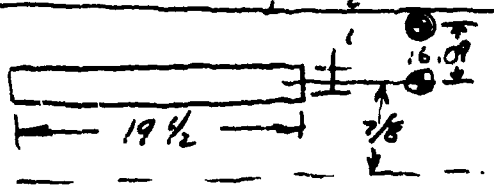
BOTTOM EXTRUSION (ONE)



$$AREA = 10 \times 3/8 = 3.75\ IN^2$$

$$I_{CG} = \frac{(10)(3/8)^3}{12} = 0.044\ IN^4$$

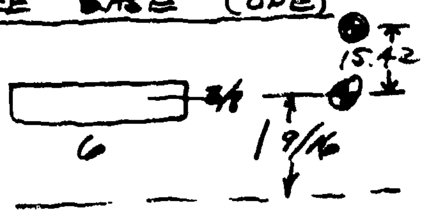
STEEL SPLINE, EQUIVALENT IN ALUM. (ONE)



$$AREA = (19.5)(1) = 19.5\ IN^2$$

$$I_{CG} = \frac{(19.5)(1)^3}{12} = 1.625\ IN^4$$

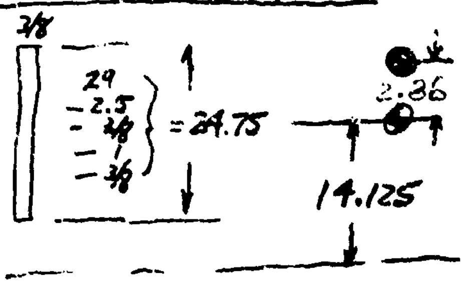
TEE BASE (ONE)



$$AREA = 6(3/8) = 2.25\ IN^2$$

$$I_{CG} = \frac{(6)(3/8)^3}{12} = 0.0264\ IN^4$$

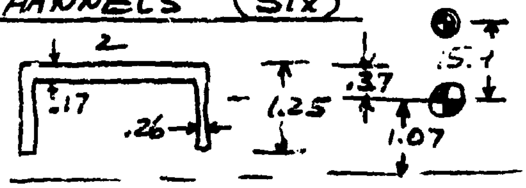
CENTER WEB (ONE)



$$AREA = 3/8(24.75) = 9.28\ IN^2$$

$$I_{CG} = \frac{3/8\ (24.75)^3}{12} = 473.8\ IN^4$$

CHANNELS (SIX)



$AREA = 2(.17) + 2[.26(1.08)]$

$= 0.9 \ IN^2$

$I_{TABLE} = 0.1 \ IN^4$

AREA C.G. ABOVE REF AXIS

A R (No.)

(48.2) (27.75)

+ (4.66) (14.25) (2)

+ (1.25) (1) (2)

+ (6.3) (.095) (2)

+ (3.75) (.1875)

+ (19.5) (.875)

+ (2.25) (1.5625)

+ (9.28) (14.125)

+ (0.9) (1.07) (6)

$\Sigma A = 96.09$

= 1632.2 = (C.G. ABOVE REF) × 96.09

∴ C.G. ABOVE REF = 16.98 = $\overline{Y}$

WEIGHT OF THESE STRUCTURAL MEMBERS:

$96.09 \ IN^2 \times 272.5'' \times 0.1 \ LB/IN^3 = 2618$ LBS FOR 1/2

OR 5236 LBS FOR BOTH

INTERIOR ROADWAYS

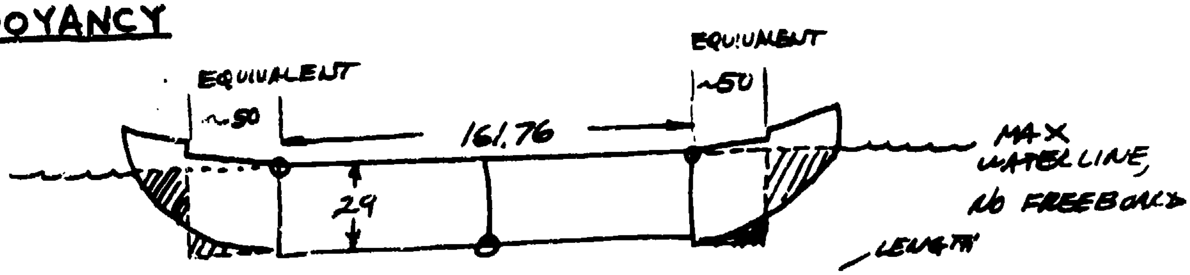
$$I_{TOTAL\ AREA} = \sum \left(I_{C.G.} + AR^2 \right)$$

DECKING	= 43.25	+	$48.2\,(10.79)^2$	5655
SIDE PLATES	+ 232.8 (2)	+	$4.66\,(2.71)^2\,(2)$	534
CORNERS	+ 1.25 (2)	+	$3\,(15.96)^2\,(2)$	1531
BOTTOM PLATE	+ 0.019 (2)	+	$6.3\,(16.87)^2\,(2)$	3586
BOTTOM EXT.	+ 0.044	+	$3.75\,(16.77)^2$	1055
STEEL	+ 1.625	+	$19.5\,(16.09)^2$	5050
TEE BASE	+ 0.0264	+	$2.25\,(15.42)^2$	535
CENTER WEB	+ 473.8	+	$9.28\,(2.86)^2$	550
CHANNELS	+ 0.1 (6)	+	$0.9\,(15.9)^2\,(6)$	137

$= 18{,}632\ IN^4$ FOR HALF OF ROADWAY

$$Z_{TOTAL} = \frac{18.632}{\bar{Y}_{MAX}} = 1097.3\ IN^3 \text{ FOR } 1/2 \text{ ROADWAY}$$

BUOYANCY

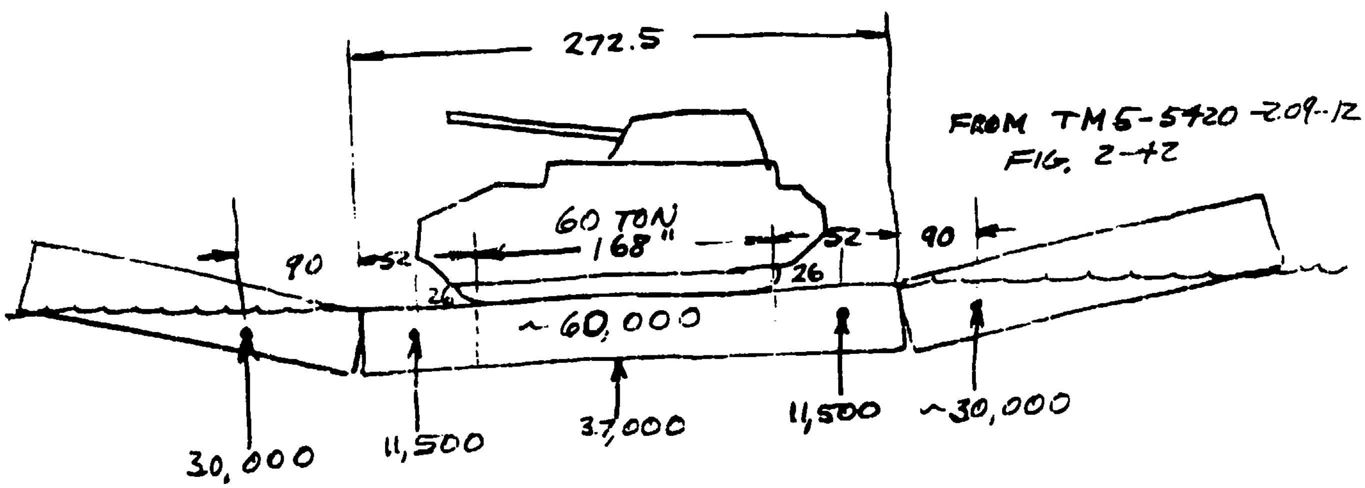


$$\text{MAX BUOYANCY} = \frac{(29)(161.76 + [2]50)(272.5)}{1728}\ 62.4$$

$$= 74{,}700 \text{ LBS}$$

$$- \text{DEAD WEIGHT} = \underline{-12{,}000 \text{ LBS}}$$

$$62{,}700 \text{ LBS NET}$$



$$\text{MAXIMUM MOMENT} = 11{,}500(26) + 30{,}000(52+90)$$

$$= 4{,}559{,}000 \text{ LB-INCHES}$$

HENCE THE MEAN NET, MAXIMUM BENDING STRESS:

$$\sigma = \frac{M}{Z} = \frac{4{,}559{,}000}{(2)(1097.3)} = 2077 \text{ PSI}$$

OTHER STRUCTURAL MEMBERS OF INTERIOR ROADWAY

CROSSWISE BULKHEADS

10 PCS × .19 × 29 × 80.37 × 0.1 = 443 LBS

END COLUMNS

$\left(3\tfrac{3}{4}^2 - 2\tfrac{3}{4}^2\right)\left([4 \times 29] + [4 \times 80.37]\right) 0.1 = 284$ LBS

$$\begin{array}{rl} 727 & \\ \times\ 2 & \text{HALVES} \\ \hline 1454 & \text{LBS} \end{array}$$

BOTH INTERIOR ROADWAYS = 1454 LBS

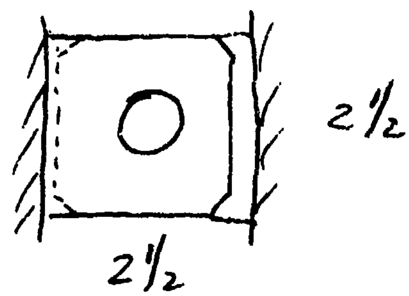
FROM SHEET 6 5236

TOTAL INTERIOR ROADWAY STRUCTURE = 6690 LBS
FOR MOMENT REACTION
IN EITHER DIRECTION

The net section stress in the Improved Wet Bridge must not exceed that in the Current Ribbon Bridge. The section depth of the Improved Wet Bridge is 9 inches greater than that of the Current Ribbon Bridge (38 vs 29 in.). Although this tends to offset the necessity for increasing its weight per unit width of roadway, the thickness of the webs must be increased because of the danger of web buckling in deep sections. Since the exact size and location of worst-case loads are unknown, the necessary increase cannot be calculated. However, from the table on p. A-2, it would be reasonable to assume that the weight of the purely structural members should increase by about 15%.

Since the load will cross the longitudinal joint between the bow and roadway pontons, the hinges must be strengthened to transfer the associated shear stress. The additional weight is calculated as follows:

ASSUME THE EQUIVALENT OF 4 CONNECTIONS EACH AT :



SOLID MATERIAL, IN ALUMINUM

EACH 275" LONG

$(2.5)^2 \times 4 \times 275 \times 0.1 = 688$ LBS

OR AN ADDITIONAL 10%

THUS THE STRUCTURAL WEIGHT WILL INCREASE BY ABOUT 25% OR

NEW STRUCTURE $6690 \times 1.25 = 8363$

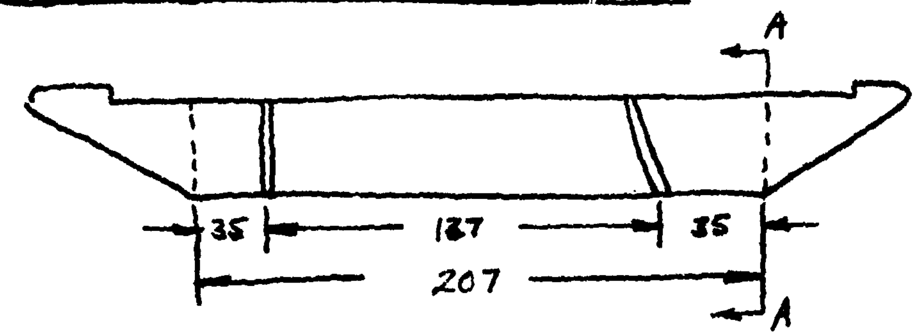
+ REMAINING NON-LOAD BEARING IS ASSUMED TO REMAIN CONSTANT $= (12{,}000 - 6690)$

$= 5310$

$= 13{,}673$ LBS

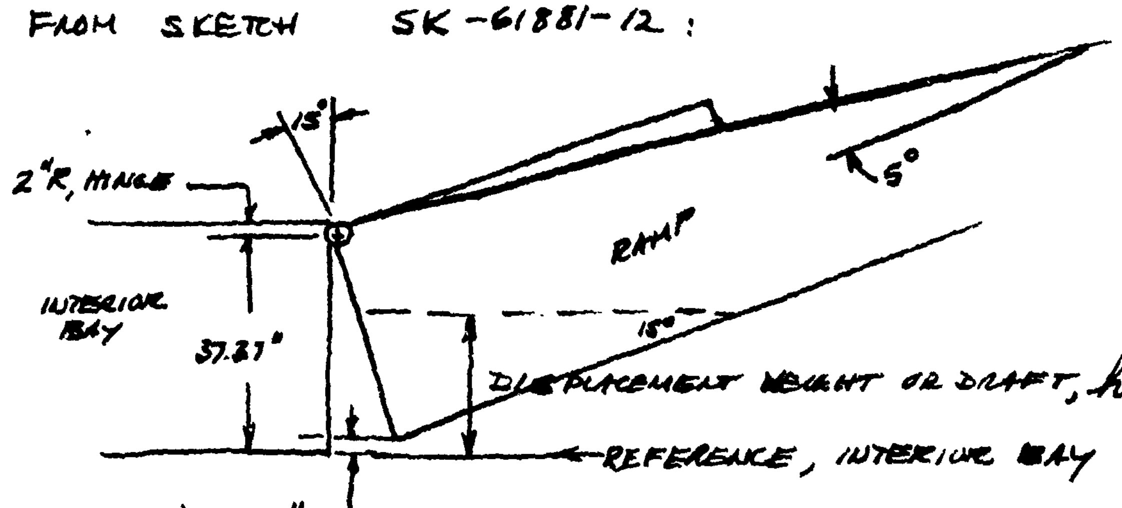
OR AN INCREASE OF 1,673 LBS

BUOYANCY OF RAMP BAY (AT 15° ONLY)



EXCLUDING SECONDARY EFFECTS, THE BUOYANCY IS MADE UP OF TWO PARTS; THE REGULAR "RECTANGULAR" VOLUME 207" WIDE, AND THE TWO "TRIANGULAR" VOLUMES (EACH IS IDENTICAL).

FROM SKETCH SK-61881-12:



$37.37 - 37.37(\cos 15°) = 1.2"$

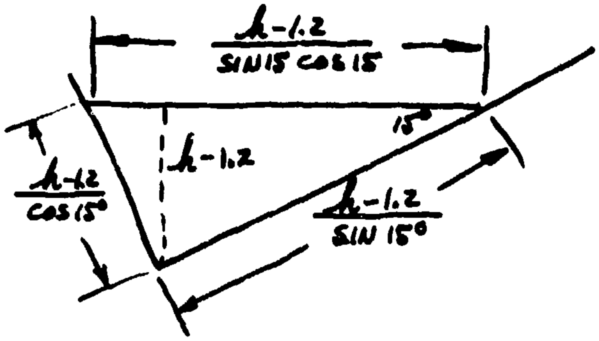
$0.9492 \quad 0.9492(\cos 15°) = .0323$ m

THE 207" WIDE VOLUME IS A TRIANGULAR PRISM.

THE TWO END VOLUMES ARE TRIANGULAR PYRAMIDS.

RECTANGULAR VOLUME AT 15°

CROSSECTION A-A



$\sin\theta\cos\theta = \frac{1}{2}\sin 2\theta$

VOLUME SHOWN IS CROSSECTIONAL AREA × 207"
5,258 m.

$$V = \frac{(h-1.2)}{2}\,\frac{(h-1.2)}{\sin 15°\,\cos 15°}\,(207)$$

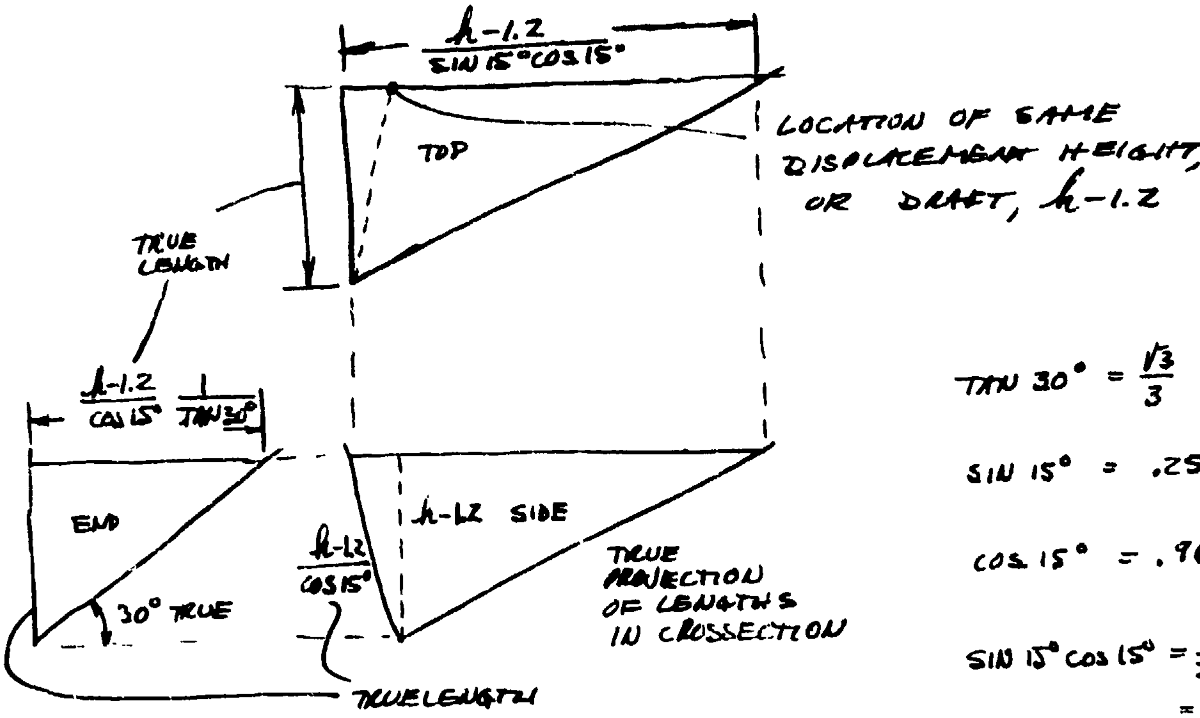
$$= (h-1.2)^2\,\frac{207}{\sin 30°}$$

$$= 414\,(h-1.2)^2 \text{ INCHES}^3$$

$$\approx 12.52\,(h-.0323)^2 \text{ m}^3$$

TRIANGULAR VOLUME AT 15°

TOP VIEW, OUTBOARD OF CROSSECTION A-A



$\tan 30° = \frac{\sqrt{3}}{3}$

$\sin 15° = .2588$

$\cos 15° = .966$

$$\sin 15° \cos 15° = \frac{1}{2} \sin (15° \times 2) = \frac{\sin 30°}{2} = 1/4$$

VOLUME OF (1) TRIANGULAR PYRAMID $= \frac{1}{3} A_{BASE} \times \text{ALTITUDE}$

$$V = \frac{1}{3}\left[\frac{1}{2}\ \frac{(h-1.2)}{\cos 15°\ \tan 30°} \times \frac{(h-1.2)}{1/4}\right](h-1.2)$$

$$= \frac{1}{6}(h-1.2)^3\left(\frac{3}{(.966)\sqrt{3}} \times 4\right)$$

$$V_{\substack{BOTH\\BOWS}} = (h-1.2)^3\left(\frac{4}{(.966)\sqrt{3}}\right) = 2.39\,(h-1.2)^3 \text{ INCHES}^3$$

$$= [illegible].39\,(h - .0323)^3 \text{ in.}^3$$

TOTAL BUOYANCY IS

$$\left(V_{RECT} + V_{BOTH\ BOWS}\right) \frac{62.4}{(1728)(2000)}$$

$$= 18 \times 10^{-6}\left(414\ (h-1.2)^2 + 2.39\ (h-1.2)^3\right)$$

$$ST = 7475 \times 10^{-6}\ (h-1.2)^2 + 43 \times 10^{-6}\ (h-1.2)^3, \ h \text{ IN INCHES}$$

$$\left(V_{RECT} + V_{BOTH\ BOWS}\right)\ 1.10\ ST/m^3$$

$$= \left[10.52\ (h-.0323)^2 + 2.39\ (h-.0323)^3\right] 1.1$$

$$ST = 11.57\ (h-.0323)^2 + 2.63\ (h-.0323)^3, \ h \text{ IN METERS}$$

THESE EQUATIONS ARE VALID FOR $h \geq .0323$.
FOR $h \leq .0323$, THE BUOYANCY IS ZERO.
WITH THE ROADWAY OF AN INTERIOR BAY AWASH, $h = 1.0$ METER. HENCE THE MAXIMUM DISPLACEMENT OF A RAMP BAY AT 15° IS

$$ST = 11.57\ (1-.0323)^2 + 2.63\ (1-.0323)^3$$

$$= 13.22 \quad \text{BUOYANCY}$$

ESTIMATED WEIGHT OF A RAMP BAY:

THE 3-PART RAMP BAY IS LONGER AT 22+ FEET THAN THE RIBBON BRIDGE AT 18+ FEET. THE 3-PART RAMP BAY IS TAPERED OVER THE ENTIRE LENGTH, WHEREAS THE RIBBON BRIDGE IS TAPERED OVER ONLY ONE HALF OF ITS LENGTH. THESE TWO FEATURES ARE OFFSETTING, SO THE WEIGHT INCREASE OF THE 3-PART RAMP IS ASSUMED TO BE THE SAME AS THE INTERIOR BAY ESTIMATED WEIGHT INCREASE, NAMELY, 15%.

DEADWEIGHT, STRUCTURE & OTHER SUPPORTING MASSES: $11,700 \times 1.15$ =	13,455 LBS
ONE EACH OF:	
6 CYLINDER DEUTZ DIESEL	961
SCHOTTEL PUMP-JET	353
PUMP-JET DISPLACEMENT LOSS	177
FUEL	130
CLUTCH, SHAFTING, STRUCTURE, ETC.	400
	15,476 LBS

OR 7.74 S T

HENCE THE NET BUOYANCY OF A 3PART RAMP AT 15° WITH THE INTERIOR ROADWAY AWASH IS $13.22 - 7.74 = 5.48$ S TONS.

IN GENERAL THE NET BUOYANCY (WHICH MAY BE NEGATIVE) IS, IN SHORT TONS:

$$S.T. = 11.57\,(h - .0323)^2 + 2.63\,(h - .0323)^3 - 7.74$$

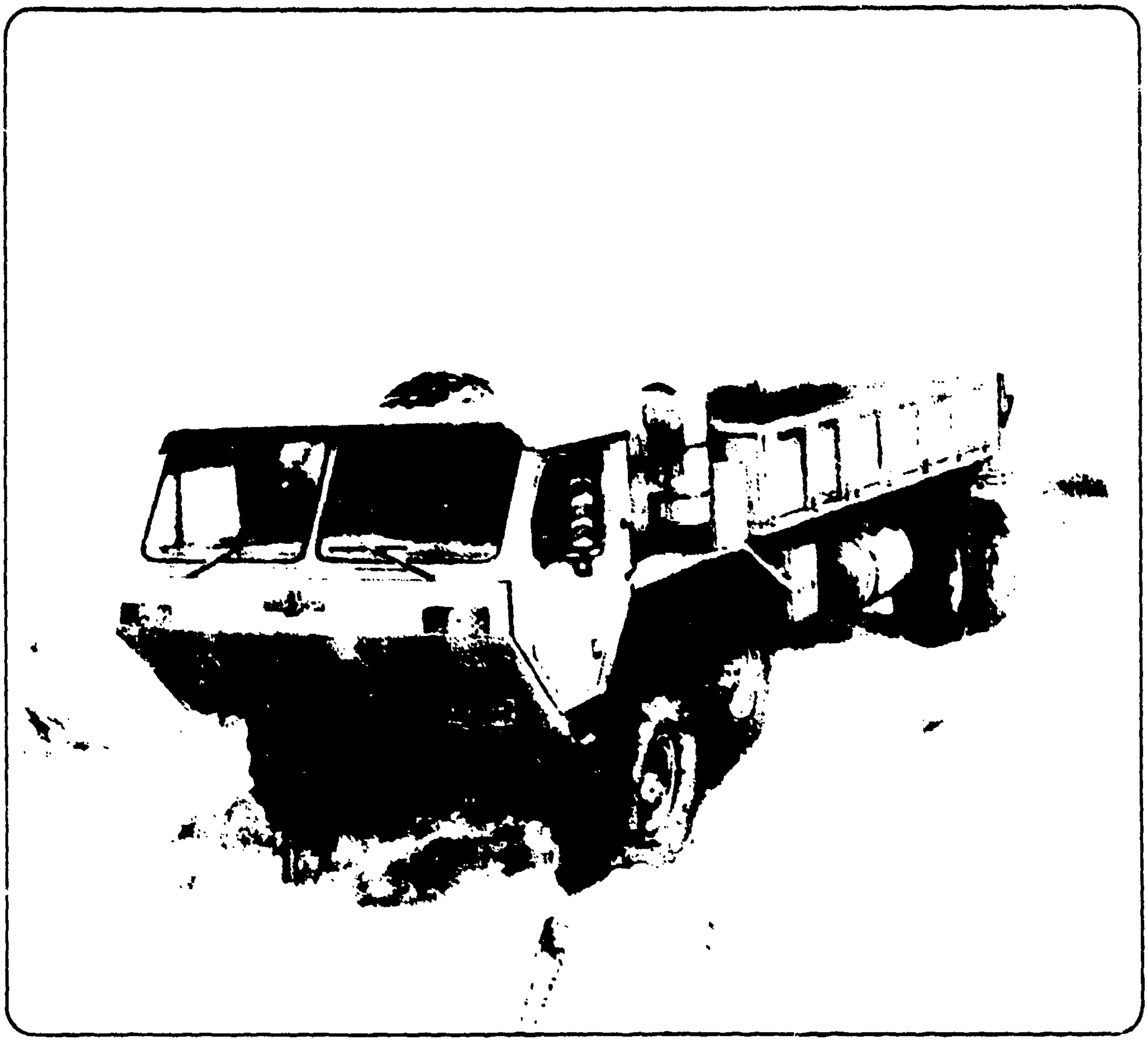
FOR h, IN METERS.

APPENDIX B

SPECIFICATIONS FOR HEAVY EXPANDED MOBILITY TACTICAL TRUCK

UNITED STATES ARMY HEMTT

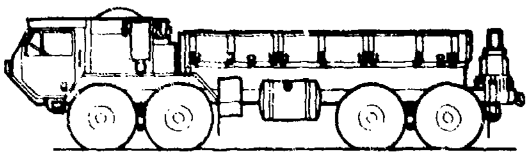
(HEAVY EXPANDED MOBILITY TACTICAL TRUCK)



OSHKOSH

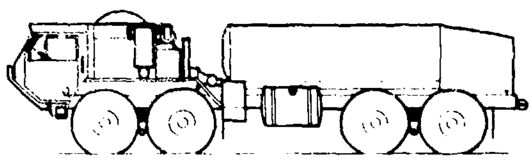
HEMTT SPECIFICATIONS

The United States Army Heavy Expanded Mobility Tactical Truck (HEMTT) is an eight wheel drive, 10 ton, on-off highway vehicle produced in five configurations:



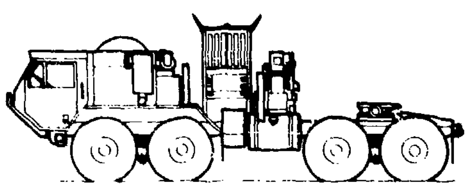
THE M977

Cargo truck with light duty material handling crane.
NSN 2320-01-097-0260 with self recovery winch.
NSN 2320-01-099-6426 without self recovery winch.



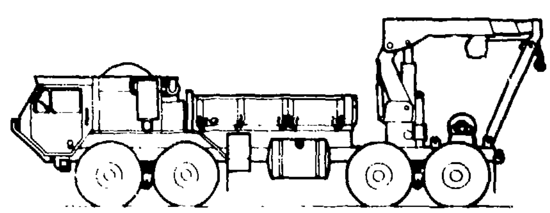
THE M978

Tanker truck used for refuelling.
NSN 2320-01-097-0249 with self recovery winch.
NSN 2320-01-100-7672 without self recovery winch.



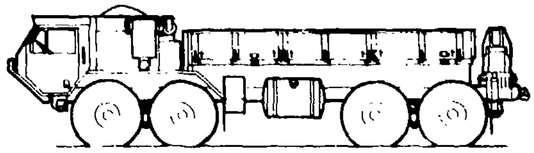
THE M983

Tractor truck with material handling crane.
NSN 2320-01-099-6421 with self recovery winch.
NSN 2320-01-097-0247 with self recovery winch but without crane.



THE M984

Wrecker truck.
NSN 2320-01-097-0248 with self recovery winch.



THE M985

Cargo truck with heavy duty material handling crane.
NSN 2320-01-100-7673 with self recovery winch.
NSN 2320-01-097-0261 without self recovery winch.

WEIGHTS & DIMENSIONS*

	Cargo M977	Tanker M978	Tractor M983	Wrecker M984	Cargo M985
Chassis Weight **	25,630 (11 533)	25,630 (11 533)	25,760 (11 592)	27,086 (12 189)	25,630 (11 533)
Front Tandem	17,539 (7 892)	17,539 (7 892)	17,514 (7 881)	17,581 (7 991)	17,539 (7 892)
Rear Tandem	8,091 (3 641)	8,091 (3 641)	8,245 (3 710)	9,505 (4 277)	8,091 (3 641)
Curb Weight	34,889 (15 700)	34,996 (15 748)	36,927 (16 617)	***42,735 (19 231)	37,189 (16 735)
GVWR	60,000 (27 000)	60,000 (27 000)	60,000 (27 000)	80,000 (36 000)	60,000 (27 000)
GCWR	100,000 (45 000)	100,000 (45 000)	100,000 (45 000)	100,000 (45 000)	100,000 (45 000)
Overall Length	396 (9 900)	396 (9 900)	349 (8 725)	378 (9 450)	396 (9 900)
Overall Width	96 (2 400)	96 (2 400)	96 (2 400)	96 (2 400)	96 (2 400)
Overall Height	120 (3 000)	120 (3 000)	120 (3 000)	120 (3 000)	120 (3 000)
Reducible Height	102 (2 550)	102 (2 550)	102 (2 550)	102 (2 550)	102 (2 550)
Wheelbase	210 (5 250)	210 (5 250)	181 (4 525)	210 (5 250)	210 (5 250)
Ground Clearance	25 (625)	25 (625)	25 (625)	25 (625)	25 (625)
Approach Angle	43°	43°	43°	43°	43°
Departure Angle	45°	45°	45°	45°	45°

* Weights are in lbs. and (kg). Dimensions are in inches and (mm).
** Deduct 1,100 lbs. (495 kg) for models without self recovery winch.
*** Deduct 9,500 lbs. (4 275 kg) for the M983 model without crane.

AXLES, FRONT TANDEM

Make and Model — Oshkosh 46K

Type — Driving, steering, single reduction, 30° front turning angle, single cardan joint, closed type steering ends.

Inter-Axle Differential — Driver controlled

Gear Ratio — 5.57:1

Rating — Maximum rated load on tires at ground 30,000 lbs. (13 500 kg)

Tandem Wheelbase — 60 inches (1524 mm)

AXLES, REAR TANDEM (All except M984)

Make and Model — Eaton DS-381

Type — Driving, single reduction

Inter-Axle Differential — Driver controlled

Gear Ratio — 5.57:1

Rating — Maximum rated load on tires at ground 30,000 lbs. (13 500 kg)

Tandem Wheelbase — 60 inches (1524 mm)

AXLES, REAR TANDEM (M984 only)

Make and Model — Eaton DS-580

Type — Driving, single reduction

Inter-Axle Differential — Driver controlled

Gear Ratio — 5.43:1

Rating — Maximum rated load on tires at ground 50,000 lbs. (22 500 kg)

Tandem Wheelbase — 60 in. (1524 mm)

BODIES/EQUIPMENT

Cargo M977	Tanker M978	Tractor M983	Wrecker M984	Cargo M985
216" (5 400 mm) cargo body 45.833 ft. lbs. (6.3 TM) material handling crane	2500 gallons (9500 liters) fuel resupply tank	2" or 3½" kingpin Universal 5th wheel 146,200 ft. lb. (20.2 TM) material handling crane	120" (3 000 mm) cargo body 120.000 ft. lb. (16.6 TM) crane 45.000 lb. (20 408 kg) recovery winch	216" (5 400 mm) cargo body 89.100 ft. lb. (12.4 TM) material handling crane

BRAKES, PARKING

Type — Spring brakes mounted on No. 3 and No. 4 axles. Modulated split type secondary emergency system.

BRAKES, SERVICE

Type — Internal shoe, dual system air operated.

Lining Size Front and Rear — 16.5 x 5 in. (419 x 127 mm) (all except M984 rear)

M984 Lining Size, Rear — 16.5 x 7 in. (419 x 178 mm)

CAB

Construction — Extra heavy duty welded steel construction with corrosion resistant skins • Two man • 96" (2 438 mm) cab width • Tinted safety glass throughout • Rear windows • Piano type door hinges with stainless steel hinge pins • (2) Rear view mirrors • Suspension driver and passenger seat • Dual sun visors • Interior light • Variable speed air windshield wipers • Windshield washers • Electric and air horns • Heater and defroster • Seat belts with retractor.

Instrumentation — Imperial/Metric gauges • Tachometer with engine hourmeter • Speedometer with odometer (miles) • Air pressure gauge • Lube oil pressure gauge • Fuel level gauge • Coolant temperature gauge • Voltmeter • Air cleaner condition gauge • Ammeter • Throttle control • Low air pressure warning • High coolant temperature warning • High beam indicator light • Transmission oil temperature gauge • Turn signal indicator lights • Blackout light controls.

CHASSIS EQUIPMENT

Front steel fenders • Extra heavy duty front bumper and skid plate • Sealed beam headlights • Cab identification lights • Stop, tail and turn signal lights with 4-way flashing front and rear • Front marker lights • Cab clearance lights • Blackout lights • Horizontal muffler • 100 gallon (380 litre) fuel tank side mounted • Stowage compartment • Spare tire and davit • External hydraulic connection • Service and emergency air brake connector - front and rear - left side • Slave start connector • Trailer electrical connector.

COOLING SYSTEM

Radiator Core — Fin and tube type.

Frontal Area — 1710 sq. in. (11,032 cm²)

Water Pump — Gear driven centrifugal type.

Construction — Fabricated top and bottom tanks and side members bolted together to form a rigid frame surrounding the radiator core. Built-in deaeration system.

Fan — 32 inch (813 mm), 8 blade dual belt driven. Temperature modulated clutch.

Transmission/converter oil cooler

ELECTRICAL SYSTEM

Alternator — 62 amp, 24 volt (All except M983)
100 amp, 24 volt (M983)

Starting — 24 volt

Lighting — 24 volt

Batteries — (4) 12 volt, 900 CCA each @ 0°F

ENGINE

Make and Model — Detroit Diesel 8V92TA

Type -- V8, two cycle

Bore — 4.84 in. (123 mm)

Stroke — 5.0 in. (127 mm)

Displacement — 736 cu. in. (12.06 liters)

***Brake HP** — 445 (332 Kw) @ 2100 RPM

***Max. Torque** — 1250 ft. lbs. (1695 N•m)

*SAE Standard Conditions — 29.38 in. Hg. (995 mbar) and 85°F (29°C)

FRAME

Type — Formed channel, bolted construction with grade 8 bolts.

Material — Carbon manganese steel, heat treated.

Yield Strength — 110,000 psi minimum (758 mPa)

Size — 10-1/8 x 3-1/2 x 3/8 in. (257 x 89 x 9.5 mm)

Section Modulus — 17.415 in.³ (285 cm³) per rail

RBM — 1,915,650 in. lbs. (216,439 N•m) per rail

STEERING SYSTEM

Type — Integral hydraulic main and booster gears.

Ratio — 23:1

SUSPENSION, FRONT AND REAR (All except M984 rear)

Make and Model — Hendrickson RT340

Type — Spring with steel saddle and equalizing beams. 10" (250 mm) vertical axle travel.

SUSPENSION, M984 REAR

Make and Model — Hendrickson RT450

Type — Spring with steel saddle and equalizing beams. 10" (250 mm) vertical axle travel.

TIRES

Type — Tube

Size — 16.00R20

Tread — Radial traction

TRANSFER CASE

Make and Model — Oshkosh 55000, two speed.

Type — Air operated front tandem disconnect.

Ratio — .98:1 and 2.66:1

TRANSMISSION

Make and Model – Allison HT740D

Type — Automatic, with torque converter, four speed.

Ratios — 4th — 1.00:1
3rd — 1.38:1
2nd — 2.02:1
1st — 3.69:1
Rev. — 6.03:1

WHEELS

Type — Steel disc

Size — 20.00 x 10.00